

Problems  
Relating to the Development of  
**INTERNAL COMBUSTION  
ENGINE INDUSTRY**  
in India



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PROBLEMS RELATING TO  
THE DEVELOPMENT OF  
INTERNAL COMBUSTION  
ENGINE INDUSTRY  
IN INDIA

Proceedings of the Symposium  
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# Introduction

The Internal Combustion Engines Research Committee at its meeting in December 1950 agreed to a suggestion that a symposium might be arranged to discuss various problems relating to the development of this key industry in India and thereby bring together representatives of Government, industry and research institutes for discussion of subjects of common interest. The Board of Scientific and Industrial Research accepted the recommendation of the Research Committee in March 1951 in this regard.

Prof. M. S. Thacker, Director, Indian Institute of Science, Bangalore, requested Dr. J. C. Ghosh, Chairman of the Committee, that the symposium be held at the Institute in the Internal Combustion Engineering Department which has now become a very active centre of research in this branch of engineering.

The Committee was grateful for this offer of co-operation, and the symposium was decided to be held at Bangalore under the joint auspices of the Internal Combustion Engines Research Committee and the Indian Institute of Science.

Tentative proposals for the symposium were circulated in December 1951 among various industrial organisations, Government departments and research institutes inviting their contributions to the symposium. A final programme was drawn up for a two-day session on 5 - 6 April 1952 in Bangalore at the Internal Combustion Engineering Department of the Indian Institute of Science. It included three technical sessions and one discussion group in addition to the inaugural session on the morning of the opening day. Messrs. Burmah Shell Oil Storage and Distributing Co. of India, Ltd., the British Information Service and the Standard Vacuum Oil Co. were kind enough to show some of their technical films in the evenings.

Over 200 delegates and invitees participated in the various sessions of the symposium. A visit was arranged on the occasion to the I.C.E. Laboratory of the Indian Institute of Science, and a separate brochure was issued by the Director of the Institute on the various research projects that were on exhibit in the laboratory. A special feature of the occasion was a test run of the Derwent Mark V gas turbine unit on the test-bed that has been erected for gas turbine instruction and research.

The papers presented at the symposium may be classified into three groups:

- (1) those relating to research and development carried out in India by the authors ;
- (2) those reviewing recent developments abroad ; and
- (3) those presenting proposals for investigation and development in future in this country.

Papers of the first category have naturally received preference and have been printed in great detail. In view of the limitation of space, however, papers of the other two groups have been summarised to indicate the line of approach the authors have in mind. It is hoped that this procedure which has been followed for reasons of economy will meet with general approval.

# Inaugural Address

Dr. J. C. GHOSH

Chairman, Internal Combustion Engines Research (I.C.E.R.) Committee, Council of Scientific and Industrial Research

I have great pleasure in welcoming you to this first symposium to be held in India on problems relating to the development of internal combustion engine industry in this country.

The symposium has been organised, as you are aware, under the joint auspices of the I.C.E.R. Committee and the Indian Institute of Science. This Research Committee came into being early in 1942 when the Government, pressed hard by the exigencies of war, became interested in the indigenous manufacture of internal combustion engines and components. An Expert Committee, presided over by Shri J. C. Mahindra, conducted a survey of the then existing industry and the possibility of improving the means of production so as to make India independent of foreign supply. How stupendous this task was can easily be judged by some of the findings of the Committee. The Committee also suggested several important research problems, like the study of high duty alloy cast irons for replacing nichrome steels, the application of producer gas plants for operating high speed heavy duty trucks, the coal dust motor and the use of vegetable oils as lubricants and fuels ; some of these are now being investigated in several of our laboratories.

The Committee was reconstituted in 1946 under the Council of Scientific and Industrial Research which has become in many ways the centre of all industrial research activity in the country. The Committee is now responsible for the co-ordination of all research work in the field of internal combustion engines, for the grant of aids to industry and laboratories for carrying out scientific research programmes of value, for advising the Indian Standards Institution on various standards developed in the field and for assisting the Indian Patent Office in their queries. The following research schemes have been approved for investigation :

1. Development of Hot Air Engines by Prof. H. A. Havemann and Shri N. Narayana Rao.
2. Development of Gas Turbine and Jet Propulsion Unit by Dr. S. R. Sen Gupta and Prof. Fairbairns.
3. Instruction and Research on Gas Turbines by Prof. H. A. Havemann, Shri N. T. Gopala Iyengar, K. Narayanaswamy and M. A. Tirunarayanan.
4. Development of Heavy Duty Parts for I.C. Engines by Precision Casting and Die Casting by Prof. M. S. Thacker, Prof. H. A. Havemann and Dr. A. Ramachandran.

5. Electropolishing for Internal Combustion Engine Components by Prof. Brahm Prakash, Prof. H. A. Havemann and Shri S. Ramachandran.
6. Design and Development of a Combustion Chamber Test Rig and Turbine Blade Test Rig for the Development of a Prototype Gas Turbine by Prof. H. A. Havemann.

Of these, the first three schemes have been in operation since 1951. The next two have just started and the last one is awaiting financial sanction of the Government. The variety of schemes that have been taken up for investigation is illustrative of problems that arise in the development of this key industry. It should, however, be remembered that it may take a few years before an idea considered feasible in the laboratory reaches the stage of design, tooling and production.

During the last three years, considerable interest has been shown in the development of the gas turbine power plant. It is true that the reciprocating engine industry itself has not yet been developed fully and it will be still some time before we can claim to have indigenous designs on production lines. All the same, the gas turbine has become such an important power plant that it has been considered advisable to start at least two main centres of research and development in this field. Some facilities for advanced instruction in the design, operation and testing of gas turbines are also now available in the Institute laboratory.

Of the various types of internal combustion engines, the diesel engine industry has become well established in India and a few firms are producing engines of indigenous design. The engines are usually of the single cylinder horizontal or vertical type, generally used as a power plant or as a drive for pumps. Most of the component parts for the engines are now made in India, but the finer components, like the injection pump and nozzle, are yet being imported. A report, however, is now being prepared by the National Physical Laboratory of India and the Secretary to the Committee on the possibilities of undertaking the manufacture of injection pumps and nozzles. The diesel engines from one firm in India have also brought to light the value of alloy cast irons of the Meehanite series. A large number of castings including such intricate and high duty parts as the cylinder head and the crankshaft have now been made of silicon cast irons. This experience in the use of high duty cast irons should prove extremely valuable in saving the use of scarce alloying elements, like nickel, vanadium, molybdenum, etc. The diesel engine industry is now looking forward to the production of high speed heavy duty transport vehicle engines which should naturally be of the multicylinder type. An important bottle-neck in this is the combustion chamber which is invariably guarded by rigorous patents abroad. It is now proposed to take up a research scheme on the development of a combustion chamber suitable for this purpose ; and it is hoped that in the next three or four years when this development may be fruitful, the Indian industry may undertake manufacture of multicylinder engines of their own design.

The automobile industry is gradually gaining ground in the country but

of the various firms who are engaged in this section of the industry, there is only one firm that has gone into the production of engines on a limited scale. It is, however, gratifying to note that there is now one complete machine shop for the machining of all principal parts of an engine and a foundry is getting ready for the preparation of all major castings. As you are aware, we are poor in modern foundry techniques and it may be some time before the art of making intricate castings is understood by our technicians.

Charcoal producer gas plants for transport vehicles were built in large numbers during the war in view of the meagre gasoline supplies. With recent developments in cyclone coal burners and pressurized gas plants, gas producers are likely to compete with other power plants in efficiency and economy. This problem should therefore receive high priority in our programme of research.

Apart from land transport and power plant engines, we have not touched even the fringe of other important fields, like marine engine, the air power plant or the defence weapons. With the development of shipping industry one should expect a sizeable demand for large diesel engines and probably at a later date for turbines. Indigenous development will have to wait till the smaller multicylinder engines are built in India and we have sound experience of their design and manufacture. In the aeronautical field, it is a pleasure to note that the Indian trainer craft HT-2, designed and built in India, is now on the air ; and we may very soon start on the design and production of a suitable power plant for this aircraft. Preliminary attempts are being made to reproduce a small aero-engine with such modifications as would suit the materials and machining facilities available with us.

In this connection I may mention two factors which should receive the attention of every one interested in the development of a large scale industry, namely, the importance of the design and development section and the inspection department. In all mass production on assembly line principles, the design and production planning offices hold the key to the success of any endeavour. One has only to see the catalogue of parts listed for the assembly of an engine to realise the enormous design and planning effort that has gone into the production of a simple engine. We often find that in India little importance is given to this aspect of production.

Standardisation and inspection play an equally important part. If the consumer is to have faith in an indigenous article, it is absolutely necessary that the product is of defined and consistent quality. From the producer's point of view, of course, the need for standardisation is essential when provision has to be made for exchange of parts from one unit to another. Hence rigorous inspection should be insisted upon at every stage of production. The variety of materials and processes that go into the making of an engine naturally demand a capable inspectorate having authority to reject doubtful products. A greater realisation of the importance of this fact should help place our industry on a much better footing and our products at a level comparable to similar products from abroad.

Speaking in broad terms about an industry of the type and magnitude

of the internal combustion engine, one might remember that the manufacture of an engine is usually achieved by a group of subsidiary industries each of which is specialising in a component part of the engine. There are, for example, factories exclusively organised for the manufacture of such parts as spark plugs and injection pumps and nozzles ; there are many subsidiary organisations for the manufacture of standardised pistons and rings and the casting or forging of crankshafts and connecting rods. Such subsidiary industries have not yet been started in India, though again it is very encouraging to note that one firm has now taken up the manufacture of pistons and rings of standardised design and specifications. In a way we are here in a vicious circle ; the components industry cannot really gain ground till large assembly units come into existence ; on the other hand, the capital and the organisation involved in the initial set up of a large assembly line are enormous, and the prospect of profits at the initial stages is not bright. All the same, the manufacture of such parts as injection units, spark plugs and gaskets is very important and should not be delayed too long. In this connection I may mention that valuable reports have been drawn up on the design and manufacture of spark plugs, pistons and rings ; and I commend them to those who are interested in the manufacture of these component parts.

This symposium today relates to the development of the internal combustion engine industry. The main problems naturally fall into two groups: (a) fuels that have to be burnt in engines and (b) materials which may be used in the fabrication of engines. Being a scientist and not an engineer, I am naturally more interested in fuels and their combustion.

The ideal engine should be able to use any kind of fuel ; in this respect the gas turbine perhaps comes nearest the ideal. While the fuel used in any engine should depend upon the type of combustion chamber in the engine, many other considerations, like the cost, weight, bulk and danger of storing in the vehicle, also exercise a significant influence on the choice of fuel for any vehicle. Same fuels, therefore, are ruled out for certain purposes, while the search for making the engine accept different kinds of combustible material continues. The major fuel for the internal combustion engine is yet the liquid fuel. About 40 per cent of the liquid fuel in use belongs to the gasoline class. Our own resources in this direction are poor. But two refineries are being started in India which should give us greater confidence in regard to a regular supply of gasoline. Petroleum refining today should not be looked upon as a unit that makes gasoline and lubricating oil only, but rather as a highly technical operation that refines and manufactures products like gaseous hydrocarbons, olefines, gasoline, solvents, fuel oils, lubricating oils, asphalts and petroleum coke, to say nothing of the numerous special products. When thermal cracking has reached what appeared to be the final stage of development a few years ago, the maximum octane number level was rather low. However, with the advent of catalytic cracking, the petroleum industry developed another tool which has made possible the production of a larger percentage of gasoline from a barrel of crude oil at a higher octane number level than could be obtained by thermal cracking.

Gasoline quality (as well as the quality of other products) is affected to a considerable degree by the characteristics of the crude oil from which it is produced. Increased sulphur content is the principal problem presented by most of the sources of petroleum. A survey of the sulphur content of several products produced from the crude oil by distillation makes it quite apparent that most of the sulphur compounds occurring in crude oil are contained in the higher boiling fractions. However, it is these heavier portions (gas oil and residuum) which are used as charge stock to cracking units for the production of gasoline, and as would be expected, the sulphur content of the cracked gasoline would be proportional to that of the charge stock from which it is made.

The second problem presenting itself in the production of gasolines is the hydrocarbon type base of the newer discoveries. The majority are paraffinic or, at best, mixed base, which emphasises the problem of obtaining good gasoline yield having the desired anti-knock quality characteristics. In general, gasoline produced from paraffinic crude oils by straight run distillation are of relatively low anti-knock quality, in some instances as low as 20 octane number. Gasolines from naphthenic crude oils, in contrast, frequently have an octane number rating in the range of 70 to 75. This same trend is noted also in the octane numbers of the gasolines produced by thermal cracking of gas oils from paraffinic and naphthenic crude oils.

This brings me to the question of high octane fuels. During the last two decades, as the demand for high powered engines increased, there has been a continuous increase in the octane numbers of gasolines used in engines. Increased power has demanded higher compression ratios and this has resulted in every means being employed to improve the octane numbers. The discovery of tetraethyl lead and other octane increasing compounds has given further fillip to this race for higher and even higher compression ratios. How far this increase should be allowed unchecked depends in a way upon the comfort and speed that are expected in passenger cars. Recently, however, there is a shift in emphasis on obtaining high compression ratios. Instead of going in for higher octane fuels, an attempt is being made at obtaining the same result by altering the design of the combustion chamber. Studies in this field call for fundamental investigations on the nature of combustion of liquid fuel particles, the movement of the flame front and the nature of chemical reaction during the movement of the flame. Recent researches in the propagation and expansion of the flame front in actual engine combustion chambers have given valuable data and led to the development of Chrysler single valve hemispherical combustion chamber which gives a compression ratio of 8.5:1 for normal octane values. It is gratifying to note that work of such fundamental value has also been started in India. In this connection, I may make a suggestion that such investigations should include the radiation of heat from flame fronts to the surrounding walls which indirectly affects combustion of fuels by variations in the temperature of the cylinder head and the adjoining cylinder wall. One would also recommend the extension of the use of injection pumps for light low viscosity fuels in the open

combustion chamber. Till now injection pumps have ante-chambers and auxiliary chambers including the Ricardo "comets". The open combustion chamber which is sometimes said to be not particular of the quality of the fuel and is generally giving a smooth combustion may perhaps be tried for petrol injection.

The next important liquid fuel is the heavier diesel fuel. The two-stroke diesel engine in some ways is the future reciprocating power plant of the world. The major advance in diesels today is not so much in the improvement in the quality of fuels as in the super-charging of these engines. Super-charging is better than the method of increasing the compression ratio because of the considerably less maximum pressures in the former case, thus permitting less heavy stress-bearing components and therefore a lighter engine.

Alcohol comes as the next major liquid fuel. Considerable importance is being attached to the power alcohol industry by the Government and before long, we may have sizeable resources on hand to make great use of this fuel. About 20 per cent mixture with motor gasoline has now come to be recognised as a suitable fuel for most vehicles. But the approach to this problem of utilising alcohol perhaps needs revision. Serious attempts should be made to design an engine primarily for running on a high percentage mixture of alcohol and gasoline. No doubt alcohols are by no means ideal as fuels as they contain unwanted oxygen in their molecule and therefore have comparatively low heat values, but for a number of reasons they have to be considered important fuels. Their importance lies in the ease with which they may be prepared, in a high degree of purity, from a wide range of raw materials, growing crops, waste organic products, such as straw and sawdust. One may summarise the main advantages of alcohol blends as (1) high anti-knock value, (2) reduction in boost temperature and (3) prevention of freezing in carburettors, and the disadvantages as (1) low calorific value and (2) increased danger of fuel stoppage due to separation.

A discussion on liquid fuels would be incomplete without a few words on kerosene. Being not subject to as high a duty as petrol (the present figure is about 3 as./gallon), superfine kerosene is nearly half as cheap as petrol. Of course it has gained exceptional value as a fuel with the advent of the gas turbine which runs mostly on kerosene when used for air transport. We in this country should be specially interested in the use of kerosene as a fuel in our agricultural machinery. The fuels used in the carburettor type tractor are gasoline, distillate and kerosene. It is encouraging to note that the majority of tractors under production in the commonwealth countries run on kerosene or diesel fuel, though corresponding American production is mainly on the gasoline class. The reason is obvious ; the Americans have abundant supplies of gasoline and distillate products. We should feel encouraged by this trend towards the use of heavier fuels and kerosene in agricultural machinery. The recurring cost of fuel which sometimes becomes equal to the cost of an engine in about 6 to 10 years of engine life really brings out the advantage of kerosene-burning engines for us in India. There are serious disadvantages like the necessity for two fuel tanks, one for starting and the

other for running, difficulties of starting and running at idle and light loads and crank case dilution. Crank case dilution necessitates more frequent changes of lubricating oil and causes excessive wear. But the basic economy factor is of such great importance that time and money spent on the development of a suitable carburettor and supply system would appear well utilised.

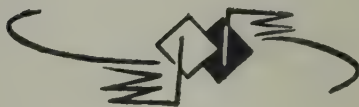
There is another class of liquid fuels which is now gaining ground in large surface transport and stationary gas turbines—the very heavy liquid fuels like tar. The problem of injection of these fuels now appears satisfactorily solved. But their combustion yet presents all the problems that one meets with in the burning of pulverised coal. Heavy fuels have to be preheated before admission to the chamber—in fact many grades could not be pumped at all without such preheating. During the process of combustion of these fuels, particles of coke and other forms of solid fuel are formed and these have to be given time to burn before being passed into the turbine. A promising solution to this problem is based on the use of a cyclone or vortex in which particles are held in a whirling air stream. As the air flows past the particles and their combustion proceeds, the particles move in towards the centre of the vortex where they are completely consumed. Then there is the problem of carbon formation and deposition on the wall of the combustion chamber. Such information as is available shows that carbon formation is promoted by over rich mixture strength in the primary zone, low air turbulence, large droplet size, increasing carbon-hydrogen ratio and increasing aromatic content of the fuel. The last three are specially characteristic of heavy fuels. Intensive work will be required in this direction to overcome these disadvantages. It is known that the introduction of water vapour greatly reduces the formation of carbon, presumably by setting up the water gas reaction, and therefore the introduction of water into compressors for reducing compressor work may have a double value. I understand there is a possibility of examining this matter in some detail in connection with investigations on cyclone producer gas plants in India and we should look forward eagerly to the data that may be collected.

Last of all, I come to solid fuels which are now being increasingly recognised as the future economic fuels for stationary power plants. We have in this country almost inexhaustible sources of low grade coal and it should be our earnest endeavour to find a power plant that can burn it economically and with high efficiency. The combustion of pulverised coal presents many problems, like ash deposition, carbon deposition, corrosion of metal chambers, etc., which I have referred to earlier. But the most interesting study is the combustion of the coal particle itself. I hope the Fuel Research Institute will take in hand this fundamental problem for careful investigation.

The conversion of low grade coal into high grade liquid fuel is now an established industrial practice in countries where the indigenous production of petroleum is meagre. Indian production of petroleum does not meet more than ten per cent of her requirements. Hence project reports were prepared sometime ago in the Directorate General of Industries for manufacture of gasoline from low grade Indian coal, at least to the extent of making us

independent of foreign supplies for our defence requirements. Last year at the session of the Science Congress in Bangalore, I gave a full account of the schemes which were under the consideration of the Government. I would, therefore, not repeat what I said on that occasion. The urgency for implementation of that project is not so great now, in view of the fact that two refineries, one handling a million ton of crude oil and the second of 2 million capacity will soon be established in India. This will, indirectly, have the effect of stockpiling crude oil which will be sufficient for a year's defence requirements. It is hoped that in the second five year plan, synthetic oil industry will be given a high priority, and will, before long, be one of the major industries of India.

I thank you for your generous response to our invitation to attend this symposium. I am grateful to the authorities of the Indian Institute of Science for making all local arrangements for this symposium. Prof. Thacker, Prof. Havemann and Dr. Ghatage are active workers in this field, whose co-operation has been extremely useful to the Research Committee. I hope that the co-operation between the Government, the Council of Scientific and Industrial Research, the Industry, and the research workers which is so evident in this symposium will extend to the development of internal combustion engine industry in India. May this co-operation lead to the establishment of a flourishing industry in the country.



# Review of Gas Turbine Research Work in the Bengal Engineering College, Sibpur, Calcutta

Dr. S. R. SEN GUPTA

Bengal Engineering College, Sibpur.

The scheme for the development of a small prototype Gas Turbine for test and performance study and finally of a Jet Propulsion Unit was sanctioned by the Council of Scientific and Industrial Research in July 1950. Preliminary study of some aspects of this problem had commenced earlier. The task of survey of literature was entrusted to a number of officers in the different departments of the College. As soon as the scheme was sanctioned, a symposium was held. It is gratifying to report that the Council of Scientific and Industrial Research have decided to publish some of these papers in a monograph.

My colleagues in the Mechanical Engineering Department (Prof. R. G. P. S. Fairbairn, Prof. D. Banerjee, Prof. B. K. Dutt) assisted by Sri S. P. Sen, a Ministry of Education senior research scholar and Sri T. Das, a research assistant on the scheme, necessarily have had to and have still to shoulder the responsibility for the major part of the work on this project inasmuch as design and production of the different components of the prototype are their responsibility. My colleagues in the Metallurgy Department (late Prof. Walter Baukloh and now Dr. G. P. Chatterjee) have been in charge of investigation on the development of suitable high temperature resistant materials. Dr. H. Rakshit of our Physics and Communication Engineering Department has been charged with the responsibility of developing instruments and appliances for measurement and Dr. S. K. Chakravarty, Head of the Department of Mathematics has been guiding the work of the Research Assistant engaged in the solution of related hydro-dynamic problems. We also have had the benefit of advice and guidance from Dr. E. Weingartner, who was for a time attached to our Fuel Technology Section, in the design of the combustion chamber. The workshop staff of the College have also been rendering valuable assistance in producing components and accessories according to our design and specification. Five junior Research Assistants were sanctioned for the scheme. But unfortunately during most of the time up to now no more than three were available.

## **Compressor**

The problem of developing the compressor was taken up first since the efficiency of the unit depends very largely on the efficiency of the compressor. Three different types of compressors (capable of delivering fairly large

quantities of air required) could be considered in this connection, viz., the Lysholm, the centrifugal and the axial flow compressor. The choice was guided by the following three considerations:

- (a) in recent years it has been found that an axial flow compressor can be made to be more efficient than others.
- (b) such a compressor can be built and tested stage by stage.
- (c) the experience we should gain in the manufacture of blades would also serve us well in manufacturing the turbine.

Taking into consideration the possible cost of manufacture, the availability of materials and workshop facilities as well as the facilities for testing we decided to develop a turbine unit which would yield an available surplus of 80 B.h.p. It was anticipated that the design and manufacture of such a small size gas turbine unit would present many difficulties. But in the circumstances and from considerations of the possibilities of such small rotary power units we feel that our choice was fully justified.

For the design of the compressor unit fullest use has been made of published results of investigations.<sup>1-5</sup>

The following are some of the important design data for the compressor:

Air inlet temperature	...	...	25°C.
Air inlet pressure	...	...	14.7 lb./sq.in. abs.
Mach No. (velocity of air divided by local velocity of sound) not to exceed	...	...	0.7
No. of stages	...	...	12
Temperature rise per stage	...	...	11.85°C.
Air outlet pressure	...	...	42.9 lb./sq.in. abs.
Air outlet temperature	...	...	162.70°C.
Mean diameter of the rotor	...	...	7.0 in. to 6.54 in. at the last stage
Blade height	...	...	0.880 in. to 0.530 in. at the last stage
Revolution per minute	...	...	22,800
Theoretical stage efficiency	...	...	78.25%
Air mass flow	...	...	3 lb./sec.
Designed shaft output of the complete Gas Turbine unit	...	...	80 h.p.
Power utilised to drive the compressor	...	...	262 h.p.
Type of blading N.A.C.A.C.4	...	...	50% reaction (both in stator and rotor)
No. of blades per stage	...	...	50
Blade chamber	...	...	Circular
Angle of twist of blades	...	...	5°
Blade tip clearance	...	...	0.02 in.
Axial clearance	...	...	0.1 in. along the mean diam.

With the choice of 50 per cent reaction blading the pressure rise in the different stages of the rotor and stator will be at the same rate.

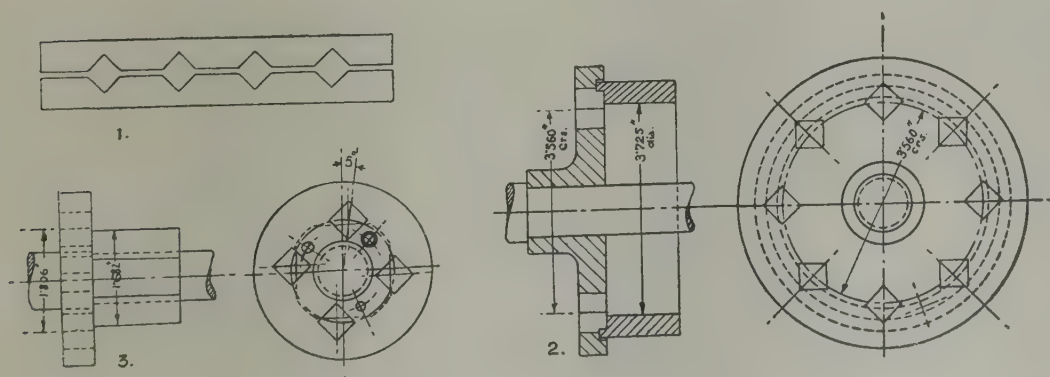
Great difficulties were experienced in the manufacture of the component parts of the compressor, especially of the compressor blades which are of aerofoil section and are required in large numbers. This task has been successfully tackled, and a simplified manufacturing method has been finally evolved. This simplified technique has been adopted even though it has entailed some slight deviation from the shape of the chosen aerofoil section. This was unavoidable since neither the College workshop nor any workshop in the neighbourhood has the costly machines which would be required to produce large number of blades of this kind. We anticipate some interesting results from the study of the compressor characteristics owing to the deviation from the true N.A.C.A. blade profile.

The blade sections at the root and at the tip are more or less identical in regard to cross-section and chord-length and the profile approximates very closely to circular arcs. It was possible to design suitable fixtures to manufacture the blades. The manufacturing technique employed may be summed up into the following operations:

(1) turning circular blanks of the correct size from bar material ; (2) milling square sections at the root ; (3) producing flat blade blank by straddle milling at the proper angle with the root in suitable fixtures ; (4) turning concave surface of blade section with specially designed fixtures ; (5) turning convex surface of blade section with specially designed fixtures ; (6) pantograph milling of the leading and the trailing edges ; (7) imparting the twist and producing the slight variations of the sections at different blade heights by means of metal dies at elevated temperature ; (8) milling root sections ; and (9) hand polishing and inspection.

Figs. 1, 2, 3 show the different fixtures used for the blades. If adoption of this method of manufacture does not materially reduce the efficiency of the compressor unit, it will prove to be an advance in the technique of manufacture of blades since blades produced by this method would be far less costly than those produced in the usual way.

Although the final unit has been designed for 12 stages, preliminary work is being confined only to the first 3 stages. The assembly of the rotor and



FIGS. 1, 2, 3—DIFFERENT FIXTURES USED FOR THE BLADE  
1. MILLING FIXTURE ; 2. INSIDE TURNING FIXTURE ; 3. OUTSIDE TURNING FIXTURE

the stator for the first three stages has been completed and it is now ready for preliminary test. Figs. 4, 5 and 6 show the compressor disc, the stator and rotor assembly and the rotor being balanced in a dynamic balancing machine.

The testing of the compressor has presented us with a further difficulty. The shaft speed of the compressor has to be about 23,000 r.p.m. which is a very high speed indeed. To begin with, a Ford V8 engine has been suitably mounted on a test bed. A gear box has been designed and constructed to increase the shaft speed to about 24,000 r.p.m. This engine will drive the compressor during the preliminary stages of testing. Already bearing and lubrication difficulties have arisen owing to the high speed of rotation involved. However, attempts are being made with some degree of success to get over this difficulty. The quantitative results of tests on the compressor are expected within a very short time. If results approximate closely to the design characteristics of the compressor, the remaining nine stages will be constructed and test of the complete compressor carried out. If, however, the efficiency is not up to the desired figure for the first 3 stages, the question of revision of design will have to be gone into.

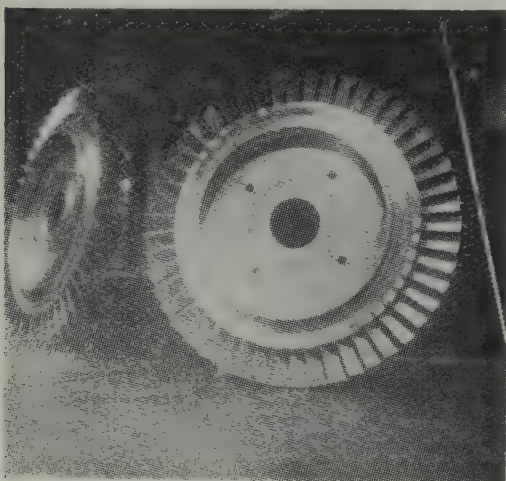


FIG. 4—COMPRESSOR DISC



FIG. 5—STATOR AND ROTOR ASSEMBLY

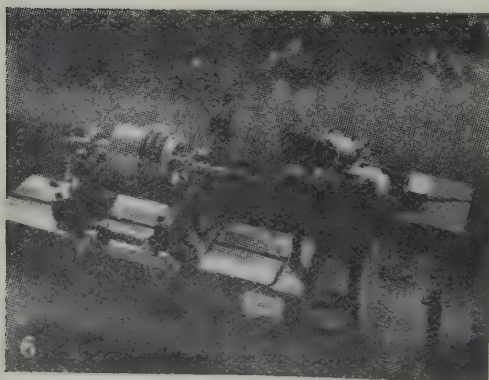


FIG. 6—ROTOR BEING BALANCED IN A DYNAMIC BALANCING MACHINE

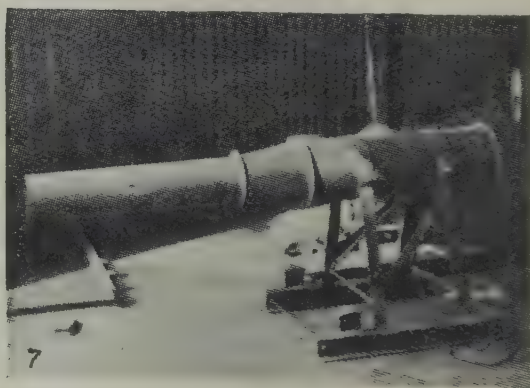


FIG. 7—COMBUSTION CHAMBER AND THE COMPRESSOR

## Combustion Chamber

The development of the combustion chamber has been taken up simultaneously. We have had for this work even less information than for the design of the compressor assembly. A combustion chamber having the following dimensions has been designed and constructed from considerations of fundamental principles:

Type of combustion chamber				Straight flow Brown-Boveri type
Air mass flow	...	...	...	3 lb./sec.
Compressor exit temperature	...	...	...	167°C.
Compressor exit pressure	...	...	...	42.5 lb./sq.in. abs.
Turbine inlet temperature	...	...	...	650°C.
Fuel flow at full load (kerosene)	...	...	...	145 lb./hr.

Fig. 7 shows the combustion chamber and the compressor and fig. 8 shows the sectional view of the combustion chamber. The Brown Boveri type of combustion chamber was chosen to ensure as efficient wall cooling as possible so that materials of inferior grade could be used. The unit being very small, a single combustion chamber will be used and will suffice for our purpose. The entry of air from the compressor to the combustion chamber has been taken at right angles to the direction of air flow. This will no doubt cause some loss in efficiency, but the other alternative, viz., to bring it in line would entail serious constructional difficulties, specially in a unit of such a small size. Pending the development of the compressor for preliminary studies a model combustion chamber is being tried out by using a centrifugal blower which delivers 0.25 lb. of air per second. The fuel is injected by an atomising nozzle from a fuel pump driven by an independent electric motor. The choice of the type of flame tube has justified itself in that the wall cooling has been found to be very efficient even when the temperature of the gas leaving the combustion chamber was as high as 645°C. After a good deal of effort it has been possible to produce fairly stable combustion. The main dimensions of the

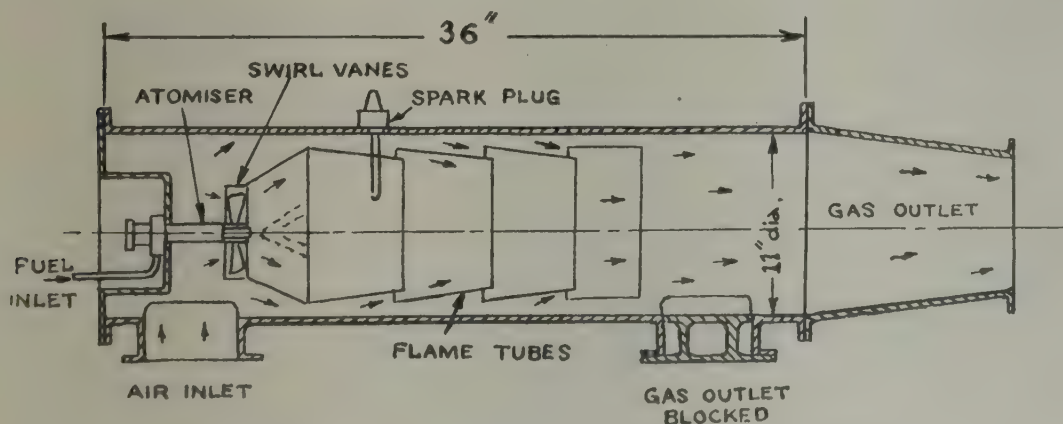


FIG. 8—SECTIONAL VIEW OF THE COMBUSTION CHAMBER

combustion chamber on which studies are being made are as follows:

- (1) Diameter of the combustion chamber ... .. 11 in.
- (2) Length ... .. 36 in.
- (3) Average air velocity in the combustion chamber before  
ignition ... .. 32 ft./sec.  
(present practice—80 ft./sec. in aircraft gas turbine  
to 30 ft. per sec. in industrial and marine installations)
- (4) Rate of heat release— $0.6 \times 10^6$  B.t.u./cu.ft./hr./atm.  
(For stationery and marine application  $(0.5 \text{ to } 1.5) \times 10^6$  for burning  
residual oils)  $0.1 \text{ to } 10^6$

The flame tube has been constructed in sections so that there may be no distortion. For stream line flow, the flame velocity is of the order of 3.5 ft. per sec. and it varies from fuel to fuel and also with air-fuel ratio. The apparent flame velocity can be increased 10 times or more with suitable arrangements for turbulence, provided flame stability is maintained i.e., the apparent flame velocity is greater than the air velocity. In order to reduce the size of the chamber the air velocity has to be high for a stable flame which, of course, means large pressure loss. In the final design, a compromise will have to be made between the size of the chamber and the allowable pressure loss.

In the present design, mixing is obtained by introducing air in annular streams parallel to the gas stream thereby obtaining an efficient wall cooling, somewhat at the expense of effectiveness of mixing. Mixing effectiveness has been sacrificed for better wall cooling because of the non-availability of high temperature resistant materials for chamber construction.

The Ford V8 engine with the high speed gear box mentioned earlier in conjunction with a turbo supercharger obtained from the Air Headquarters through the kindness and courtesy of Air Vice-Marshal S. Mukherjee will soon supply requisite quantity of air for low pressure testing.

### Turbine

A two-stage turbine to be linked with the compressor and the combustion chamber has been designed and some progress has been made for its construction. The dimensions of the turbine are as follows:

Mean diameter of the rotor	...	...	...	7.75 in.
Type of blade	...	...	...	50% reaction at mean diam.
Blade height	...	...	...	0.9 to 1.25 in.
Total power to be developed	...	...	...	353 h.p.
Gas temperature at inlet	...	...	...	650°C.
Gas outlet temperature	...	...	...	442°C.
Adiabatic stage efficiency	...	...	...	85%
No. of blades	...	...	...	55
Blade tip clearance	...	...	...	0.04 in.
Axial clearance	...	...	...	1/8 in.

In view of the low temperature (650°C.) for which the turbine unit has

been designed, it has been decided to use 18-8 (Cr-Ni) stainless steel for the blades and 13 per cent nickel steels for disc construction.

It has been suggested that we might attempt to use hollow blades in the turbine and use ordinary high carbon steels, Nimonic materials being difficult to obtain. We have, however, decided as mentioned above, to use stainless steel since a comparatively low temperature is involved and since we are also of the opinion that the construction of hollow blades presents such technical difficulties as we are unable to solve at the present moment. For the present we are aiming to develop a turbine unit that is capable of yielding the requisite h.p. and one that will be required to work for short periods. When suitable materials have been developed then only we should be in a position to construct a unit that would be required to do sustained work at a high temperature.

### Development of Materials

Investigation has been in progress for the development of ceramic blades using suitable metal-ceramic mixtures. When we take into account the fact that the operation temperature is of the order of  $1000^{\circ}\text{C}.$ , oxidation resistance on prolonged exposure, low density and therefore less centrifugal stress at the high speed are some of the advantages of use of ceramic materials. To obtain high ductility, resistance to creep, dimensional stability, and resistance to spalling it is necessary to carry out investigation on the effects of the different variables like relative proportion of different materials, degree of fineness, sintering temperature etc., of the mixture. Iron and alumina, also iron and silica mixtures, compressed into the desired shapes under a steel die, have been sintered in hydrogen atmosphere in an electrical resistance furnace at about  $1000^{\circ}\text{C}.$  for 3 to 4 hrs. Fig. 9 shows the furnace which has been developed in our laboratories for this purpose. Some of the results are shown in table 1 and plotted in figures 10 and 11. It may be observed from figures 10 and 11 that Brinell Hardness numbers of the specimens decrease with the silica per cent for small periods of sintering but increase with the silica per cent for longer periods. Thus there may be a minimum critical period for sintering before the requisite strength, hardness and allied properties

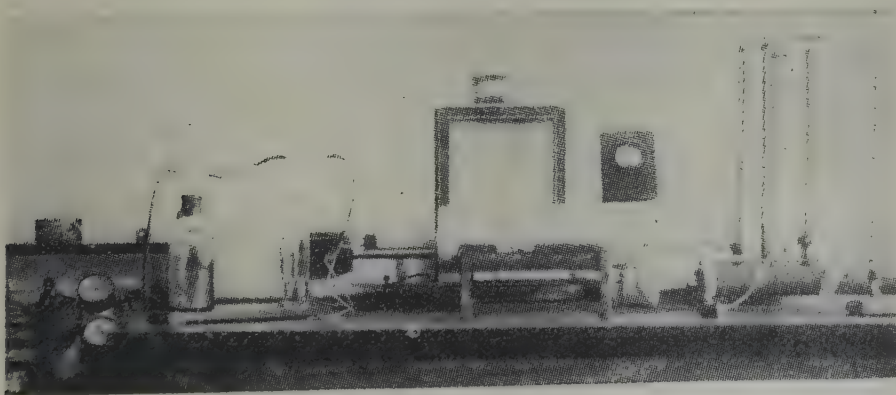


FIG. 9—FURNACE SET UP FOR THE PRODUCTION OF CERAMIC BLADES

TABLE 1—COMPOSITION BY VOLUME OF 170-MESH 'METAL-CERAMIC' MIXTURES AND THEIR MECHANICAL PROPERTIES AFTER SINTERING AT  $1020^{\circ} \pm 5^{\circ}\text{C.}$  FOR 6 HRS.

COMPONENTS	COMPOSITION BY VOLUME %	COMPRESSIVE STRENGTH TONS/SQ. IN.	B.H.N.
Iron-Alumina mixture	Fe—50		
	Al <sub>2</sub> O <sub>3</sub> —50	5.4	—
	Fe—60		
	Al <sub>2</sub> O <sub>3</sub> —40	11.3	—
	Fe—70		
	Al <sub>2</sub> O <sub>3</sub> —30	19.0	—
	Fe—80		
	Al <sub>2</sub> O <sub>3</sub> —20	23.0	—
Iron-silica mixtures	Fe—90		
	Al <sub>2</sub> O <sub>3</sub> —10	—	23.8
	SiO <sub>2</sub> —50	—	74.1
	SiO <sub>2</sub> —40*	15.4	50.3
	SiO <sub>2</sub> —30	—	42.4
Iron-Magnesia mixtures	SiO <sub>2</sub> —20	—	56.8
	MgO—50	3.0	—
	MgO—30	2.98	—
	MgO—20	—	15.0

\* The mixture was sintered at  $1000^{\circ}\text{C.}$ ; linear shrinkage, 2.7% and vol. shrinkage 7.9%.

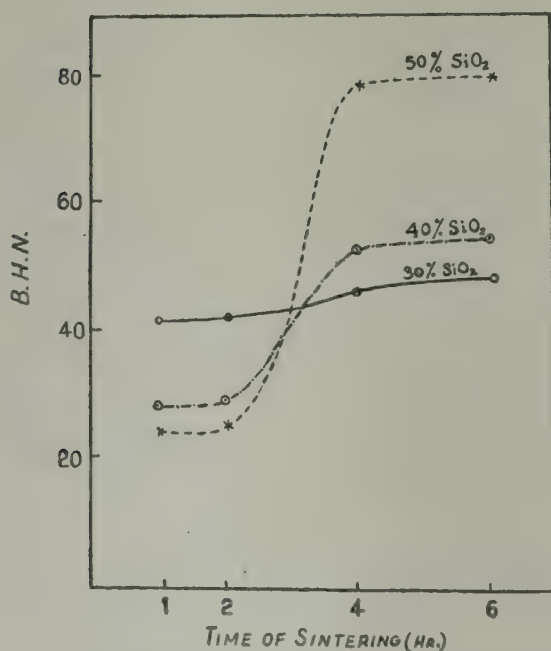


FIG. 10—VARIATION OF BRINELL HARDNESS WITH TIME OF SINTERING

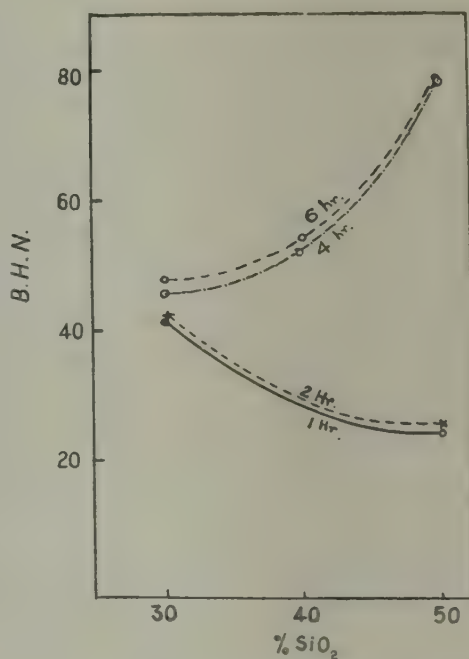


FIG. 11—VARIATION OF BRINELL HARDNESS WITH PERCENTAGE OF SiO<sub>2</sub>

may be obtained. Higher percentage of silica greater than 50 per cent. tends to cause devitrification with poor resistance to thermal shock and creep.

Recently more encouraging results have been obtained with a mixture of iron and kaolin powder. Attempts are also being made to test spalling and shock resistance properties of these ceramic materials. Work has also been taken up to develop a high creep resisting alloy steel using elements which are available in the country.

### Measuring Devices

For testing both the compressor and the combustion chamber measurements of static and total pressure, temperature, the mass flow, the speed and torque are presenting special problems and instruments are being designed and constructed. For measurement of inter-stage pressure of the compressor, a specially designed cylindrical pitot tube has been constructed. In addition to the routine work of construction and calibration of instruments like thermocouples, pitot tubes, etc., work has also been done for the development of an electromagnetic pickup device for accurate measurement of high speed of rotation. A toothed iron disc mounted on the shaft develops an alternating e.m.f. measured by comparison with a calibrated audio-generator on a cathode-ray oscillograph (Fig. 12). The accuracy of frequency measurement and hence r.p.m. by this system has been estimated to be 0.25 per cent.

For measurement of torque it has been decided to adopt some electronic methods. The phase angle between the e.m.f.'s developed across two pick up coils, as in the measurement of r.p.m. by two identical toothed discs with a known separation on a shaft is obviously a measure of the torque. A tentative phase meter capable of measuring small phase angles was designed and tested with sinusoidal voltages. The preliminary results have been encouraging but calculations have shown that its performance is dependent upon the harmonic contents of the two e.m.f.'s. The development of a new type of phase meter for measuring small phase angle is also under consideration.

### Aerodynamic Calculations

Attempts are also being made to predict by calculation the aerodynamic

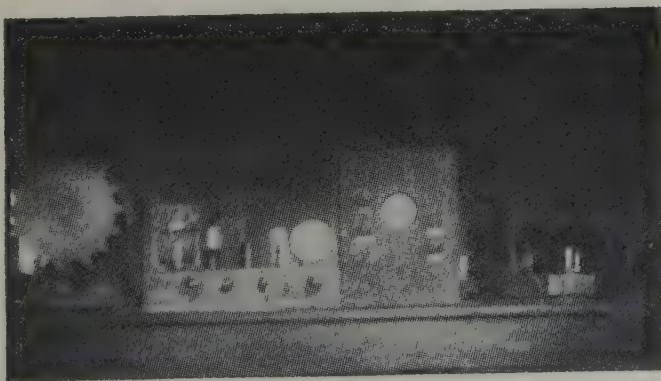


FIG. 12—DEVICE FOR THE ACCURATE MEASUREMENT OF HIGH SPEED ROTATION

characteristics of cascade of blades of the profile which has actually been used in the compressor.

### Acknowledgment

I am greatly indebted to the Council of Scientific and Industrial Research for sanctioning this scheme and providing for it five posts of junior research assistants and some money for the contingent expenditure. I, however, very much regret that owing to the poor scale of salary for these posts we have hardly had more than three research assistants working on the project at one time. I have also to thank the Government of West Bengal for providing a sum of Rs. 22,000/- as capital expenditure for the initial stage of the development work. To Dr. J. C. Ghosh, the chairman of the Internal Combustion Engines Research Committee, I am deeply grateful for the sustained interest he has been taking in this research project and for giving me the opportunity of reporting on behalf of my colleagues the progress of our work.

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# Review of work carried out in the Internal Combustion Engineering Laboratory at the Indian Institute of Science, Bangalore

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This is the first occasion for the Department of Internal Combustion Engineering to appear before the public and to give an account of the work it has done so far.

The general aim of the work of the Department is based on the present state of the internal combustion engine industry in India and the economic and social needs for which it can provide a remedy. The scope of the present activities can be said to be the following.

Design of new types of engines to suit Indian materials: fuels and lubricants will be developed and prototypes will be produced. All possible experimental tests will be carried out to ascertain the performance of the prototype engines, so as to encourage the industry to take over the manufacture on a large scale.

Researches of fundamental and applied nature will be conducted on conventional engines and on engines using fuels and lubricants available in this country.

It may be mentioned that this conforms to the terms of the Tata Trust upon which the Indian Institute of Science was founded and it must be stressed that this is a development which is unique in India in the all important field of engineering.

I shall now consider in greater detail some of the specific problems which the Department at present is striving to solve.

The internal combustion engine, as we all know it from any car or motor bicycle, requires, for obvious reasons, a very clean fuel. The fuel burns inside the engine and it thus comes into contact with the moving parts of the engine. This imposes severe conditions, especially in regard to the nature of the products of combustion, the ash content, and the general chemical composition of the fuel and in fact, these conditions prevent the utilization of low grade fuels which are abundant in India. If use could be made of these low grade fuels, a problem of great national importance would be solved: it would not be necessary to spend large amounts of revenue on imports of high grade fuels.

The type of engine which would be suitable to accept those fuels and thus would have to be independent of the individual qualities of the fuel depends on the magnitude of the output required. For small output, the hot air engine offers the ways and means for a possible solution of the problem. For high output, the gas turbine, the new-comer in the field of

prime movers, is just about ready to be applied under those conditions as mentioned.

Under the auspices of the Council of Scientific and Industrial Research, a research scheme on 'Hot Air Engine' is undertaken on, roughly, the following lines. The heat engine uses air as the working medium to which heat is added from hot combustion gases through an external heat exchanger. The system has been investigated theoretically and calculations have been made to determine the thermodynamic properties and the performance of the cycle, and especially to find out the most suitable design of the heat exchanger. Efforts are now being made to complete the installation of the test equipment for which a detail design has been made. The installation of the prototype is nearing completion.

I may underline, however, that although the limitation on the quality of the fuel is practically removed, a new limitation has come in: the thermal efficiency. I personally think, however, that if the engine would really come up to the expectations, it is of small importance how efficiently it solves the problem. We will also have to keep in mind that the engine does not require any costly injection equipment nor is it necessary to install any electrical equipment.

In the field of higher output, I have already mentioned that the gas turbine offers great advantages. In the so called "closed cycle" of the gas turbine the quality of the fuel is of hardly any influence and therefore, in India, emphasis should be laid on the development of a closed cycle gas turbine prototype. This aim, I must admit, is very high indeed, and I feel it is wise to approach it in steps and to develop first, a gas turbine prototype of small output. By doing so, we will also learn how to use to best advantage, the facilities available in this country. This will at the same time give ample opportunities for the training of technical personnel in this branch of power plant engineering.

In this Department, therefore, emphasis is laid, for the time being, on conventional gas turbines to use indigenous fuels, whether in pulverized form, or in a gasified form. In view of the abundant supply of low grade coals in India, work has been started on the efficient utilization of these fuels in pulverized form. A suitable device for the continuous injection of pulverized fuel into pressurized combustion or gasification chambers of gas turbines has been designed so that it could be adopted for gas turbines later on. An injection apparatus in the form of an air-ejector, based on orthodox design principles, was manufactured, and with the knowledge obtained from experiments on this, a new and original design has been evolved which is being tested. The new ejector has been successful in overcoming considerable back-pressure.

Difficulties of using low grade fuels internally in an engine can be overcome to some extent by externally gasifying the fuel and by the introduction of a cleaning process in the flow of the producer gas before it enters the secondary stage in the combustion chamber of the power plant.

The process in itself, though not new, has up till now the disadvantage of being very bulky and costly, and slow in operation. An investigation has

indicated that a cyclone chamber has special advantages, such as low weight, small bulk, and high rates of output when used as a chamber to gasify heavy liquid fuels and pulverized solid fuels. Designs for an experimental set-up are ready for using heavy liquid fuels in the first instance, for both primary and secondary zones.

These methods will be applicable to gas turbines as well as to reciprocating gas engines and it is here that the problem of converting a petrol engine to utilize gaseous fuels attains considerable importance.

Two lines of approach are possible. One is to use the original fuels in lumps, and this can be done in common gas producers, which are at present the subject of investigations. In the alternative, the fuel can be gasified in pulverized form, a technique which will make use possibly of the cyclonic gas producer, as already mentioned.

A test bed for the testing of common producer gas plants has been constructed. For this a six cylinder Chevrolet engine has been installed with a 50 h.p. dynamometer to measure the power output. The same dynamometer is being used as a motor for driving the engine, if used as a suction unit, to enable the study of the characteristics of fuel beds, such as pressure loss, etc., under varying conditions of flow. A test cone has been developed. The deficiency in power output, amounting to a 50 per cent fall in power, as compared to the use of petrol, is the chief drawback of converted petrol engines using producer gas. The Roots blower used to precompress the gas air mixture is at present driven electrically but will be coupled to the engine, mechanically or pneumatically, by providing an exhaust gas turbine. It is intended at a later stage to work the whole plant at elevated pressures by the use of suitable blowers so as to overcome this deficiency.

The simplest method of increasing the power output of a petrol engine running on producer gas, with minimum alterations, however, is to increase the compression ratio by making changes either in the cylinder head or in pistons. Experiments on these lines are being conducted on a Chevrolet six cylinder truck type of engine. A gas plant manufactured in India is erected for the producer gas fuel.

So far I have dealt with subjects which are rather unorthodox and which constitute advances in a field of engineering science which is still unknown.

The activities of the Department on the application of indigenous materials for the manufacture of engines are on less unorthodox lines. It is obvious that the quality of the materials used for any diesel or Otto engine, will influence its basic design features. It is, however, tempting to introduce some novel features when a new design is being evolved.

I want to mention first the development of a small U-type two-stroke diesel engine. The U-principle has been a great success in petrol engines, for instance, in racing motor cycles. On similar lines, a diesel engine is being developed, though there are inherent difficulties. Certain important aspects of the problem, especially the quality of scavenging, and the general technical feasibility of a U-type two-stroke diesel engine having two parallel cylinders and a common combustion chamber in one unit have been theoretically

investigated. In the light of these investigations, the design of a U-type crankcase scavenged two-stroke diesel experimental unit has been undertaken. The engine is being so designed that, with very few alterations, it can be converted into a blower scavenged U-type two-stroke diesel engine. The design of the engine is completed, and also the part drawings. The engine will be ready within a few months.

In the course of this endeavour to simplify current engine design, the Department has given a great deal of attention to low pressure injection for diesel engines. It is well known that the disadvantage of any diesel engine is the complicated and expensive injection equipment and the possibility of avoiding it is attractive from the point of initial capital outlay, simplification, etc. The principle employed is the utilization of differential pressure existing between the main combustion space and the pre-combustion chamber to force a jet of fuel into these spaces. The design for incorporating this principle into an existing single cylinder two-stroke engine is complete. An Australian QBM 2-stroke diesel engine (4 h.p.) has been chosen for experimental work. A cylinder head with a suitably designed combustion chamber which can utilize the differential pressure as mentioned has been constructed. Several types of nozzles have been tried. Though several short runs could be made, it is found that considerable difficulties may still be encountered.

A problem of similar nature is that of fuel injection in Otto engines. This has been proved during the war to be successful from the point of fuel economy and greater specific output when applied mainly to aero-engines. It has special advantages for small two-stroke engines and the Department has actively considered this aspect of the technique.

In the field of two-stroke petrol engines, the inherent difficulty is to maintain low fuel consumption. Injection of fuel into the cylinder, after the scavenging period is over, to avoid loss of charge is considered with the special view of eliminating, later on, the high pressure injection pump. The injection pump offers special difficulties for manufacture in India due to the high precision and the high price involved, combined with high sensitivity against impurities.

The injection of petrol into a two-stroke engine has been proved to give considerable improvement in fuel economy and has been accomplished in other countries so far by high pressure injection equipment which is, however, costly, complicated and hardly economical for conditions here. The purpose of the present investigation is to develop a suitable low pressure injection system for the conventional two-stroke engine. To attain even better economy, it will be applied also to the U-type engine, where stratification of the charge may be accomplished in this arrangement with better possibilities of success in fuel economy. Preliminary theoretical studies have been completed and a spray chamber is nearing completion for studying the characteristics of low pressure fuel sprays.

Designs are in progress for a model of a U-type engine for studying the flow pattern of the scavenging air. The Department has secured a U-type petrol two-stroke engine from Austria, which will be equipped with

this new injection device. Later, a three cylinder radial engine will be built which may find useful application in automotive and possibly even in the aircraft industry. The engine will also be provided with a novel system of scavenging and cooling which is still under preliminary consideration.

In the field of four-stroke engines selected (J. A. P. air-cooled 250 c.c. 4-cycle petrol engine) the flow pattern was determined with a variable mixture carburettor. A special test rig for the injection part alone was fabricated in the Department and fitted with the available diesel fuel injection pumps and injectors, and a study of the atomization of petrol was made with the help of a stroboscopic method, with respect to various cam rates, injectors and injection pressures. The results were helpful in the process of selecting the best possible injection settings.

The injector along with the necessary accessories was installed on the engine and satisfactory results have been obtained. The engine ran even on lean mixtures. According to the information available, this is the smallest engine ever to have run on petrol fuel injection.

The results of studies concerning the applicability of indigenous materials to internal combustion engine designs will provide a considerable knowledge in the subject of heavy duty materials so that the Department felt that a completely new design could be evolved which would play an essential role in the development of a self-sufficient Indian internal combustion engine industry. One such major project is the design and development of a 160 b.h.p. six-cylinder, in line, inverted, air-cooled, Otto engine with fuel injection for aircraft propulsion.

Consistent with ideas sponsored jointly by this Department and Dr. Ghatage, Chief Designer, Hindusthan Aircraft Ltd., with the co-operation extended by Dr. Kothari, Scientific Adviser to the Ministry of Defence, Government of India, this project has been in progress since May 1950. The work is being done by the Scientific Officer deputed by the Defence Science Organisation, Ministry of Defence. It takes specially into account the use of Indian materials and intends to incorporate general improvements, especially means to minimize fuel consumption, and to improve cooling. The proposed engine of 160 b.h.p. is eventually meant to be the propulsion unit for the H.T.2 basic trainer, designed and built by Hindusthan Aircraft Ltd. At present the D. H. Gipsy Major 10 is used and would be replaced as soon as the project leads to satisfactory results.

The scheme is split into two broad phases, viz.,

- (a) Design, fabrication and development of a single cylinder engine of 25-30 b.h.p.
- (b) Adaptation of the single cylinder to a smaller four cylinder, and later to a six cylinder version of 160 b.h.p.

Designs and type drawings of most of the major components are under preparation.

In general, the design has presented numerous unusual problems and many recent developments for improving the performance have been incorporated, viz., (1) the best positioning of valves, sparking plugs in the spherical dome

of the combustion chamber and (2) direct fuel injection and locating the injector at the apex of the spherical dome, etc.

The Department is also concerned with current engine designs to make them better suited for Indian conditions. This applies mainly to the use of heavier fuels in high speed diesel engines. This could be achieved by improving the atomization of the fuels, for instance, by the introduction of pre-heating. It may also be possible to achieve a considerable saving in foreign exchange by blending some indigenous fuels with the conventional fuels used in diesel engines.

The Department also undertakes what has been termed 'Industrial Schemes'. This work is undertaken on request from Government or private organisations and consists in testing internal combustion engines. Ten different types of engines have been tested so far. The electrical dynamometers and water brakes of the Department can test engines up to 250 h.p. The Department has also undertaken investigations for private industry on the effects of blended fuels of the heavier type and on the wear of engine components, and is conducting at present a 2,500 hrs. long endurance test.

You will have noticed that so far very little has been said about fundamental researches. In fact, the Department does far more applied than fundamental research work at present and it is with a certain feeling of regret that I make this statement. It is, however, understandable and must possibly be accepted under the present circumstances that the immediate applicability of a technical solution alone decides its usefulness for the country. But one should keep in mind that only the fundamental researches will, in the long run, provide the flow of new ideas and the tools to put them into practical realisation.

I shall now enumerate a few such problems of research which are under active consideration, without, however, mentioning their ultimate aim and application.

The influence of turbulence on the rate of detonative combustion of propane-oxygen mixtures has been investigated previously. In order to extend the work to regions of weaker mixtures, which is an interesting problem from the point of fuel economy, a combustion tube has been prepared for the measurement in this region of mixture strength. The work contemplated, at a later date, for different ratios of length: diameter of the tube has been completed for one  $l/d$  ratio, and so far the results indicate that turbulence has some influence also on the weak side of the mixture. Work on a tube of a large  $l/d$  ratio had to be postponed for lack of personnel, but instrumentation is being planned to measure the flame velocity and its distribution in the tube. Preparatory work is in progress in regard to the effect of gas oscillations on the propagation velocity of the normal flame.

One of the foremost problems in rotary engine technique is the reduction of bulk and cost of heat exchangers. The efficiency and economy of most modern power plants, such as the gas turbine and the hot-air engine, depend entirely on the heat exchanger. Theoretical work has been initiated to approach the problem of the effect of pressure fluctuations on the rate of

heat transfer. The work will be carried further in the near future, and an experimental solution appears to be more promising since the mathematical implications are very complex.

Wear of internal combustion engines depends very much on the cleanness of the air aspirated into the cylinder which in turn depends on the effectiveness of the filters. Apart from the separating effect, the filter should also have a low pressure loss so as to avoid a drop in the output of the engine. The problem has considerable importance to India on account of the high amount of solid matter suspended in the air. An investigation, therefore, has been started on separation of particles from fluid flows. A patent based on theoretical and practical considerations has been worked out in this subject and a number of tests have so far been conducted with a test set-up. Two filters, one of foreign make and another of indigenous design, are being tested for the purpose of technical comparison.

I have referred on several occasions to the new comer in the field of prime movers, the gas turbine. I have already underlined also its capability of dealing with low grade fuels. Although at present the gas turbine cannot be said to have a higher efficiency than other prime movers, I am convinced that, with the further development of heat exchangers, it will become an important power plant. We have also to be careful not to overlook other advantages the gas turbine offers, by focussing our attention too closely on the consideration of thermal efficiency alone. The gas turbine very often is much simpler and cheaper and requires less space and weight than a comparable prime mover of conventional type. For some branches of power plant engineering, the gas turbine is entirely indispensable and I may mention in this connection the aero gas turbine without which high speed flight would be impossible. In areas which are devoid of water, the gas turbine would be the most suitable power unit. Last but not the least, the gas turbine is most probably the prime mover which will be called upon to convert atomic energy into mechanical energy. The Department has, therefore, concentrated a great part of its activities on the initiation and introduction of gas turbine technology.

The following two test rigs have been installed.

(a) This Department has received a Rolls-Royce Derwent V turbojet engine through the good offices of the Ministry of Supply, London. Installation of the Derwent jet propulsion unit has been completed. The fabrication of instrument panels, jet pipe, defectors and other structural work has also been completed. The gas turbine has made several short runs.

(b) Development of a test rig for gas turbine combustion chambers. This test rig is meant primarily for the testing of normal combustion chambers for pressure loss, blow-out limits and temperature distribution. Materials to replace the high temperature resisting steels will be developed and investigations will be carried out to explore the effect of more efficient cooling methods. A compressor has been installed specially for this rig. The test rig has been designed to accommodate a gasification chamber also.

The Department has taken up the design of a gas turbine prototype. The costs of testing equipment and of the prototype as a whole would not be

too high but on the other hand, the size and the output of the prototype is such as to make it interesting for potential users in this country, i.e., for rail traction, and for stationary and industrial applications and possibly for the Defence Services. It should also be possible to derive from this prototype a turbo-jet unit to be used in aircraft. The prototype will be built as far as possible from materials available here and with the means of fabrication at our disposal. Essential parts of the equipment, such as, a water brake for very high speeds, a step-up gear for the compressor test bed, have been ordered and the driving machinery for this has been acquired from Disposals.

I shall now very briefly outline the educational activities of the Department at present.

The Department is offering at present post-graduate training and research facilities to students leading to the Associateship of the Indian Institute of Science. For its personnel, the Department provides possibilities of training which cannot be had easily elsewhere. For research work, as such, hardly any trained personnel can be found. The Department is at present in a position to train about six research students.

The research students are given a course of advanced lectures for a period of two terms in all major branches of internal combustion engineering so that a clear background of fundamentals of piston and rotary type of engines can be acquired. In addition to the lecture classes, they will be engaged in practical work on engines, in dismantling several types of engines and making observations on and measurements of the various basic components. After a certain period spent in this manner, the students receive, for at least one more term, more specialized lectures. By this time, the students will have found their field of interest so that they can be given specific problems for research work. In some cases, a problem of design or an investigation is given to them collectively before they start individually. They also receive training in special testing techniques before actual research work is started. Facilities for practical training in the internal combustion engine industry are also offered to students.

In considering the work done at the Department at present, one has to keep in mind the nature of the work in basic and applied engineering sciences. Not much has been done in India in this respect so far, and it is therefore not surprising that the amount of money necessary for conducting these activities and the time required for their successful conclusion are very often underestimated. It is my pleasant duty to put on record the understanding of the central authority of the Institute and especially of Prof. M. S. Thacker, the Director of the Institute. I take this opportunity of assuring him that the Department will make the best possible efforts to use these means for the good of this country.

# Address

PROF. M. S. THACKER

Director, Indian Institute of Science, Bangalore.

Technology is too vast for any one person or group of persons to assimilate and knowledge is a gigantic aggregate assembled by the adhesive forces of partnership. In every organisation there already exists much valuable information on work that has been done or is under investigation which may not be known outside the department in which it originated. Such knowledge is a valuable asset and if correlated intelligently with other sources of information, it is capable of being developed into a practical service which could reasonably fulfil the essential functions of design or production—both of which it could serve. All stand to benefit from an exchange of knowledge and information. To this symposium, sponsored with such an object, I have great pleasure in extending to you all a hearty welcome on behalf of the Indian Institute of Science and it is hoped from these small beginnings would emerge a larger and inter-related organisation in the not too distant a future.

Gentlemen, the advent of the Internal Combustion Engine has hastened the progress of industrial development in all countries of the western world. For the first time an efficient, light and, therefore, mobile source of power was made available which resulted in far-reaching improvements in methods of agriculture, transport and industry. The mechanisation of agriculture and the opening up of new and faster trade channels on land, sea and in the air increased immeasurably the exchange of goods and thus increased the wealth and prosperity of those countries which devoted their energy to the technical development of the then newcomer. Indeed, it would be no exaggeration to say that the modern miracles of the automobile and the aeroplane which have contributed most towards better communication, exchange of ideas, improvement in trade and commerce on an international level would not have been possible but for the invention of the light and compact power unit that the Internal Combustion Engine is.

An important advantage of the Internal Combustion Engine is that its economy is practically independent of its size. This is in contrast with the steam plant, the fuel economy of which increases with the size. The maximum fuel economy of an internal combustion engine is reached in small units, while the economy of steam movers is at its maximum only in very large plants. The Internal Combustion Engine is well adapted for small plants and particularly suitable to automobiles, farm uses and airplanes on account of the ease with which it is handled.

At present, there are in India at least 0.3 million automobiles and trucks on the roads, some 10,000 motor boats, a 1,000 aircraft in air lines and more than a 100 diesel locomotives, not to mention the innumerable units—big

and small, power, industrial and farm machinery of all descriptions. The Internal Combustion Industry is still in an embryonic stage in this country. And, as anticipation of the future is the key to adequate planning for the best use of our national resources, it is necessary that attempts should be made to keep pace with the ever-increasing demand for these versatile little power plants. Our requirements, so far, have been met almost wholly by imports from abroad.

So closely interrelated is the mechanism of modern civilization that a change occurring in one part of an industry will produce a vast number of changes in a great variety of fields. The changes call into being new occupations, services and industries. Particularly impressive are the effects on agriculture, communication, aviation, metallurgy and other allied industries with the growth of the Internal Combustion Industry.

The automobile industry as existing at present in the country is devoted to assembling of various components imported from abroad. There are a few manufacturers of diesel engines of the industrial type. Even here some of the components have still to be imported. It is absolutely essential in the interests of the nation that the Internal Combustion Engineering industry is made self-sufficient to cope with domestic requirements. Such a major development is closely interlinked with the simultaneous expansion of other vital industries such as the Iron and Steel Industry, Machine Tools Industry, etc., and the starting of subsidiary industries for the manufacture of various accessories such as carburettors, fuel injection equipment, ignition and other electrical apparatus. The production of high grade materials to withstand the severe temperatures and stresses occurring in Internal Combustion Engines and light alloys for aircraft manufacture to-day, is almost exclusively a matter for highly specialised industries and should receive top priority when an Internal Combustion Industry is planned.

Imports from various manufacturers all over the world have resulted in a wide variety in the sizes and types of engines used in the country to-day. Consequently, stocking of many different kinds of spare and equipment for maintenance and repairs is necessitated at a considerable investment. To speed the tempo of modern life, it is worthwhile for the country to concentrate on the production of engines in standard sizes and types to suit the different classes of service. This would result in a considerable saving in the manufacture of component parts and accessories.

It is quite evident that the backbone of all technical development is the item of research. This research not only includes the activities of special laboratories but also those modifications which an inspired plant operator makes. The principal product of research is the formulation of more and more problems than we have ever had before. Industries must be put on a truly scientific basis so that we shall not only have more products but infinitely better ones and at very much lowered cost to the public. Importance of co-ordination between industry on the one hand and research organisations on the other cannot be overemphasised.

The design of every Internal Combustion Engine is linked up with and

based on the quality of materials. Certain specifications must be met and it is essential that high duty steels and alloys are made available in the country. Viewing the market as a whole, no manufacturer has come forward to consider the production of high duty steels and alloys which are at present imported. He is not sure that it represents a market item that can be counted upon. If it could be proved that an engine built of Indian materials works successfully and efficiently, a large scale enterprise is bound to result.

Another item of increasing importance is the acceptance of indigenous fuels, as far as technically possible and economically advisable, for Internal Combustion Engines. This will force development and activity in rural areas where the supply of ordinary fuels is difficult. India possesses huge resources of low grade fuels and it should be of immense economic value if machines are developed to accept their use. All these aims call for intimate application of basic research to engineering science.

This Institute, I am confident, is eminently suited for these creative activities. Professor Havemann has already reviewed in detail the work that is being done in the Internal Combustion Engine branch of the Institute and with the active co-operation of the various other departments, I am sure, that substantial results will be achieved ere long.

In this field, Internal Combustion Engineering for instance, the Department of Metallurgy would be able to assist in the production and testing of heavy duty materials for crankshafts, temperature-resistant alloys for gas turbine rotors and blades and light alloys for pistons ; the Aeronautical Engineering Department in evolving aerofoil shapes, blade terms for gas turbines and compressors and the Power Engineering Department in the design and testing of electrical equipment used in Internal combustion engines. The Heat Transfer Section of the Power Engineering Department which is already working in close co-operation with the Internal Combustion Engineering Department could devote itself to problems relating to heat exchangers of all categories of importance to gas turbine power plants. The Chemistry and Chemical Engineering Sections would be helpful in the researches on the utilisation of fuels and lubricants of indigenous origin and possible development of new synthetic fuels and lubricants. Fundamental problems relating to combustion could be investigated in the Physics Department and the Electrical Communication Engineering Department would be helpful in the manufacture of engine indicators and electronic apparatus which are indispensable for the development and testing of Internal Combustion Engines.

With the installation of a few more instruments in the Department of Internal Combustion Engineering, it will be possible to carry out performance tests in turbo-jet engines and the design project on gas turbines now on hand holds out promise of application to aircraft, rail traction and other industrial purposes.

The Institute has always been ready to offer co-operation to industrialists in the solution of their problems. The successful prosecution of the projects now envisaged involve considerable finance and it should be sought from the industrialists who will be benefited in the long run. Government could

certainly be expected to assist in this development. It is also apparent that this industry would contribute materially to our national strength and resource in times of war. It is felt there is need to lay stress on the development of major industries like steel, automobile, aircraft, machine tools and power plants in general in the present national five-year plan.

Technical innovations respond to scientific analysis. Progress comes through scientific improvement in materials and through the skill with which fundamental principles are applied to the solution of engineering problems of particular projects. It is on technology alone that we can found our hopes for a high standard of living. If we, as a nation, are to march forward, it is absolutely essential that we expend more and more effort upon obtaining information which is necessary for an advanced civilisation. This is extremely important in the problem of conserving of natural resources not only for the convenience of this generation but also for the absolute necessities of the future.



## Design and Development including manufacturing processes

The first session was held on 5 April 1952 at 2 p.m. in the auditorium of the Department of Internal Combustion Engineering with Dr. S. R. Sen Gupta, Principal, Bengal Engineering College, in the chair.

Eleven papers were read and discussion followed the reading of each paper.



# Current Practice and Future Trends in the Design of Automotive Petrol Engines

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## Introduction

The present paper deals with a comparative study of the major design features of automotive petrol engines produced in England, America and the continent of Europe. It is hoped that this article will provide the Indian designers of engines with useful information on such important aspects of engine design as compression ratio, rotative and piston speeds, specific output, tractive effort, etc.

U.S.A., U.K. and the continent of Europe are three principal countries that produce automotive engines. The local conditions in these countries and those in the countries where these engines are used are taken into consideration in each case before finally deciding on the design details.

## Production in U.S.A.

Production in the United States is almost exclusively for home consumption (Export 5%) and accounts for nearly 75 per cent of the world's output in automobiles. It is a vast continent with a network of first class roads stretching from coast to coast. It has large resources of raw materials and fuel within its frontiers. The standard of living is high. These factors have created a demand for automobiles of a large size. This kind of automobile requires a large engine. Details of a typical engine of this kind are given in Table I.

There is thus a demand for a very large number of engines of high output. This has made the engine suitable for mass production at low production costs. This is the secret of the low price of American cars in spite of the fact that they are large, comfortable and fitted with many refinements.

In the past, the fuel consumption of such engines was high but in recent years much attention is paid towards obtaining lower fuel consumption. One such move is a modification to the combustion chamber and valve disposition with a view to raising the compression ratio. A second one is the 'Texaco' process of fuel injection in which the fuel is injected direct into the engine in a manner similar to the timed fuel injection in diesel engines, though the charge is still spark ignited. The authors of this process claim that the engine running on this principle can run on any kind of liquid fuel at a compression ratio as high as 10:1. A third move is to increase the compression ratio in the engine under part throttle condition so as to obtain at all throttle openings the design compression ratio and thus obtain maximum fuel economy under

TABLE 1—AMERICAN AUTOMOBILE ENGINES

No. OF CYLS.	BORE X STROKE MM.	CUBIC CAPACITY LITRE	R.P.M.	C.R.	PISTON SPEED FT./MIN.	MAX. B.H.P.	B.H.P./CYL.	B.H.P./LITRE	B.M.E.P. LB./SQ.IN.	TRACTIVE DUTY CWT. MILES/HR./B.H.P.
6-8 V8	80 x 100	3.5	3,600	6.8	2,400	100-120	16-20	26-30	100	18-21
	100 x 115	5.5	3,800	7.5	2,600					

TABLE 2—BRITISH AUTOMOBILE ENGINES

GROUP	No. OF CYLS.	BORE X STROKE MM.	CUBIC CAPACITY LITRE	R.P.M.	C.R.	PISTON SPEED FT./MIN.	MAX. B.H.P.	B.H.P./CYL.	B.H.P./LITRE	B.M.E.P. LB./SQ.IN.	TRACTIVE DUTY CWT. MILES/HR./B.H.P.
Large	6	70 x 110	2.5	4,000	6.5	2,600	80	13	22	95	18
		80 x 120		5,000	7.5	3,000	to	to	to	to	to
Medium	4	80 x 110	1.5-2	4,000	6.5	2,500	120	21	42	120	35
				5,500	7	3,100	50	12	28	100	23
Small	4	60 x 90 and lower	1.1-2.5	4,000	6.5	2,400	70	15	35	110	30
				5,000	7	2,400	40 and lower	7 to 9	25 to 35	80 to 100	26 to 30

all engine running conditions. A fourth method is to inject water or methanol into the engine at full throttle full load conditions so as to suppress the tendency towards engine knock.

All these trends are, however, still in the experimental stage and deserve to be watched with interest.

### **Production in U.K.**

Production in U.K. is in recent years aimed more towards exports (Exports 75%). In addition, many raw materials and fuels are imported. The country is small and in spite of good roads it is not possible to drive at high speeds on account of traffic congestion and short distances. The taxation system in the past provided for a tax proportional to the bore diameter of the engine. All these factors have influenced in producing an assortment of automobiles and engines each of which satisfied a particular type of use. However, it may be said that in general the British engines have longer strokes and higher rotational speeds as compared to American designs. In view of the fact that fuel has to be imported and the standard of living is lower than in U.S.A., the accent on production in U.K. is more towards economy, utility and modest comfort. These objects have been achieved by classifying engines into three groups: (i) large engines to power luxury cars, buses and trucks, (ii) medium engines to power smaller cars, station wagons, delivery vans and (iii) small engines to power the small austerity car used by people having marginal purchasing power. Table 2 gives details of such engines.

So far as the large engine is concerned there is very little difference between this and its American counterpart except that the rotational speed and piston speed are higher. The stroke/bore ratio and the B.M.E.P. are also slightly higher. These factors compensate for the lower cylinder capacity and and small engines are more or less scaled down versions of the large engine.

and small engines are more or less scaled down versions of the large engine.

Taking advantage of the system of taxation in recent years, the designers in England are also designing new engines with smaller stroke bore ratios. A notable example of this kind is the use of engines having a stroke bore ratio of 0.96 only in the latest cars "Consul" and "Zephyr" manufactured by the Ford Company of Dagenham.

From the point of view of changes in basic design there is very little to report from England except that a thorough study of petrol injection is being made by a number of companies. But results obtained to date indicate no special overall advantage in adopting this costly system.

### **Production on the Continent of Europe**

The raw material, fuel position and road conditions on the continent are as critical as in England. Economic conditions are also not very favourable to the production and use of high powered cars. Just as in England the accent in design is more towards economy and utility coupled with modest comfort. The engines produced here can also be classified as large, medium

and small groups. There is, however, on the continent an extra small group not produced in U.K. or U.S.A. Table 3 gives details of these groups.

From the Table it will be seen that the first three groups are similar to their English counterparts except that they are of shorter stroke. A close study of some recent small engines reveal a trend of design which is not found in U.K. or U.S.A. For example, the Porsche, Panhard and a few other engines are air cooled and are mounted direct over the driving axle. They are of the horizontal 'boxer' type which appears to lend itself to air cooling much better than the inline vertical engine. In certain cases, both the cylinder barrel and the piston are made of aluminium alloy, the barrel being chrome plated to resist wear.

The smallest engine group is the austerity type and has a two-stroke twin or single cylinder engine with moderate rotational speeds but reasonably good power output. This engine is also air cooled and mounted on the driving axle. This class of engine propels a very light chassis which can accommodate 4 people and has road speeds up to 45 m.p.h. Road holding is quite good and fuel consumption is 55-60 miles per gallon.

One of the recent developments in Germany is a two-stroke petrol engine with self-ignition. This engine has a variable compression head with the help of which compression ratios of 100:1 and more can be obtained. While starting it is necessary to have a very high C.R. of the order of 100:1 so that the compression temperature may reach the self-ignition value required for petrol mixture. But, while running, the order of C.R. is from 13-16. This engine can run on low octane fuels. The engine has a capacity of 166 cc. and develops 0.75 h.p. to 1 h.p. at about 6,000 r.p.m.

Another notable development in Germany is the production by Mercedes Benz of a medium powered car propelled by a diesel engine. The engine has a capacity of 1.77 litres and develops 40 h.p. at 3,200 r.p.m. It is very economical since it uses high speed diesel oil which costs half as much as petrol and the car has been priced at the same value as any petrol engined car of comparable size.

Another development, again in Germany, is the successful application of petrol injection on a commercial scale to a two-stroke engine "ILO" with the result that the BMEP is increased by 50 per cent and is of a value comparable with an average 4-stroke engine. Also the fuel consumption is very much lower than with carburation. The torque characteristics are better than a 4-stroke engine.

### **Tractive Duty**

The specific tractive duty has been taken as:

$$\frac{\text{wt. of car in cwt.} \times \text{max. top-gear speed in m.p.m.}}{\text{max. developed B.h.p.}}$$

In general the specific tractive duty on an American engine is about  $\frac{2}{3}$  that of British or continental engines. As a result the American car has good acceleration and smooth running ; involves less gear changing and has considerable reserve power. Except on the open road it runs always at part throttle

TABLE 3—CONTINENTAL AUTOMOTIVE ENGINES

GROUP	NO. OF CYLS.	BORE × STROKE MM.	CUBIC CAPACITY LITRE	R.P.M.	C.R.	PISTON SPEED FT./MIN.	MAX. B.H.P.	B.H.P./CYL.	B.H.P./LITRES	B.M.E.P. LB./SQ. IN.	TRACTIVE DUTY CWT. MILES/HR.
Large	6	70 × 90	3	4,000	7·8	2,500 to 2,800	80-100	13-19	36-40	100-120	19/27
Medium	4	70 × 75	1·25 to 2·25	4,000 to 4,500	6·5	2,100 to 2,600	40-60	11-13	28-38	90-120	29/32
Small	4	70 × 70	0·1 to 0·5	3,000 to 4,000	6·7	1,600 to 2,000	20-30	8-11	28-38	80-100	32/40
Extra small	2	70 × 70	0·5 to 0·7	3,500 to 4,000	6	1,800 to 2,000	20-18	9	30 to 40	55 to 60	40

TABLE 4—INDIAN AUTOMOTIVE ENGINES

GROUP	NO. OF CYLS.	BORE × STROKE MM.	CUBIC CAPACITY LITRE	R.P.M.	C.R.	PISTON SPEED FT./MIN.	MAX. B.H.P.	B.H.P./CYL.	B.H.P./LITRE	B.M.E.P. LB./SQ. IN.	TRACTIVE DUTY CWT. MILES/HR.
Large	6 or V <sub>8</sub>	90 × 90 90 × 90	3·6 4·7	3,600 3,600	7 7	2,200 2,200	96 128	16 16	25 25	105 105	25 25
Medium	4 6	80 × 80 80 × 80	1·6 2·4	4,000 4,000	7 7	2,100 2,100	50 50	12·5 12·5	31 31	100 100	28 —
Small	4 2	75 × 65 75 × 75	1·1 ·7	4,000 3,500	6·5 6	1,700 1,750	40 23	10 11·5	36 83	120 61	35 40

Two-stroke  
crankcase  
scavenged

and unless the designer has taken good care to provide for part throttle fuel economy, the fuel bill of an American car will be high.

On the other hand most of the British and continental engines have a high specific tractive duty and have therefore to run at near full throttle conditions and the designer must take care to provide for maximum fuel economy at full throttle conditions and normally this is the case.

### **Production Methods**

In spite of the fact that the annual production of automotive engines in America runs into millions, there are not more than about 20 individual types. On this account it has been possible to mass produce each type of engine, and thus keep down the production costs. On the other hand in U.K. and the continent, except for a few firms like Morris, Austin and Ford, where production is on a mass scale, everywhere else one finds batch production of engines. Consequently the British and continental engines cost more per h.p. A second undesirable feature is that within one class of engines there are as many as 20-30 types. In England, for example, there are 30 engine types in medium class of engines with little difference in performance and quality and still having minute differences in the sizes of different parts. Consequently, such engines can be produced only by batch production methods at a high cost to the maker as well as to the user.

### **Indian Requirements**

Although a few firms have plants in India for the assembly of cars, buses and trucks from imported components, in the real sense there is no automobile industry in the country. Experience in this country is limited to the use on roads of every type of automobile produced in U.S.A., U.K. and the continent of Europe. We have to import 9/10 of our fuel requirements and our economy and living standards are such that we should necessarily have the three groups of automobiles as in U.K. and the continent.

The figures in Table 4 have been obtained by selecting the best engine in each group after a careful study of a large number of engines in use to-day. They are based on the best current practice which tends towards shorter piston stroke, moderate piston speed, higher compression ratios and a shorter crankshaft with moderate rotational speed. The six types of engines listed in the Table will meet all the requirements of the different fields of application to be encountered in this country.

It is suggested that production of the above types be taken up on well proved orthodox lines for a start. After the industry has been stabilised any reorientation can be given on the basis of experience. On the other hand, research should be directed towards developing a two-stroke multicylinder engine with low pressure petrol injection and air cooling, taking into account as much as possible, the qualities of indigenous materials. Such a power unit will definitely be the cheapest to produce and to maintain.

# Diesel Engine Industry in India

P. L. KUMAR

Ministry of Commerce & Industry, New Delhi

In India the requirements for Internal combustion engines are chiefly limited to (i) petrol engines for automobiles, (ii) diesel engines for industrial and agricultural uses and (iii) kerosene and gas engines for air compressors, pumps, etc.

The development of petrol engines in India is of recent date. Its necessity has been felt in connection with the development of the automobile industry. Messrs. Hindusthan Motors Ltd., Calcutta, who have commenced progressive manufacture of Hindusthan 14 cars in agreement with the Nuffield Group of U.K., have already undertaken the manufacture of engines for the Hindusthan 14, in phases. They have advanced considerably from the initial stages and are at present machining the imported castings of engine blocks and forgings for crank shafts, connecting rods and many other components in their own factory. They have now set up a modern foundry and forgeshop to become independent of import of castings and forgings also.

Our largest demand of internal combustion engines is for diesel engines. Their popularity is attributed to their ruggedness and their high thermal efficiency, which is reflected in the low specific fuel consumption. The comparatively low cost of diesel fuel is also a contributory factor.

## History of the Industry

Diesel engines were manufactured in India before World War II, but the production was negligible and spread over comparatively small and unorganised workshops, excepting two units, viz., Messrs. Cooper Engineering Ltd., Satara Road, Satara, and the Oriental Engineering Works Ltd., Lahore. Some jobbing factories in Lahore manufactured sub-standard engines which found market only because of their low initial price. A firm in Kanpur commenced manufacture of 'Bharat' engines in about 1930, with a number of imported components, but this venture was short-lived.

The Oriental Engineering Works Ltd., Lahore, were the next to take up the manufacture of diesel engines in 1933. They have shifted to Shahdara-Delhi, after the partition of India. Messrs. Ruston Hornsby (India) Ltd., Bombay, and Messrs. Kulko Engineering Works Ltd., Kolhapur are also manufacturing horizontal type diesel engines.

Messrs. Kirloskar Oil Engines Ltd., Kirkee, were the first to set up the manufacture of diesel engines in the post-war period. They have entered into an agreement with Messrs. British Oil Engines (Exports) Ltd., London, for the manufacture of diesel engines of which patents, licenses, or agreements are held by the following companies of Messrs. British Oil Engines

(Exports) Ltd. (i) Messrs. Oil Engines (Coventry) Ltd. (ii) Messrs. Petters Ltd. (iii) Messrs. K. & H. McLaren Ltd. and (iv) Messrs. Mirrlees Bickerton & Day Ltd. Messrs. Kirloskar Oil Engines Ltd., are the first manufacturers of vertical diesel engines in India.

### Imports

Imports of diesel engines have been very heavy. India has expended over Rs. 25 crores in five years from 1946/47 to 1950/51 in the import of these engines as is shown under "Oil-Engines" in the accounts relating the Sea and Air-Borne Trade and Navigation of India. The import figures of engines, together with their values are shown in Table 1. Imports are made from almost all the countries, where Diesel Engines are manufactured, and in a survey made in 1951, it was revealed that over 170 different makes of engines are being imported.

It is evident from the steadily increasing imports in the post-war years that diesel engines are becoming more and more popular. And from figures collected from the Collectors of Customs, Bombay, Calcutta and Madras, it is revealed that engines up to 10 h.p. are the most in demand and those between 11 to 20 h.p. are next in popularity (Table 2).

TABLE 1—NUMBER OF ENGINES IMPORTED INTO INDIA ANNUALLY TOGETHER WITH THEIR VALUES

	No. of Engines	Value in Rs.
1937-38	2,957	54,56,149
1938-39	2,553	43,48,239
1939-40	2,140	40,92,325
1940-41	1,256	27,12,179
1941-42	1,029	22,54,275
1942-43	696	15,09,882
1943-44	536	15,89,725
1944-45	338	11,77,838
1945-46	1,393	45,93,168
1946-47	3,202	1,37,21,325
1947-48	11,699	3,50,76,712
1948-49	16,152	4,68,82,450
1949-50	37,174	8,64,23,263
1950-51	35,571	6,83,13,144
1951-52	72,365	14,73,34,052

TABLE 2—SIZES OF ENGINES IMPORTED

	Up to 10 h.p.	From 11 to 20 h.p.	Above 20 h.p.
1949	19,579	8,788	3,216
1950	21,250	8,670	3,014
1951	45,219	4,117	6,513

### Indigenous Manufacture and Production

(a) *Present manufacturers*—There are five organised diesel engine manufacturers, who are receiving assistance from the Central Government and whose production statistics are maintained. There are other small manufacturers, but their production is somewhat irregular and negligible. Four of the five organised factories are located in the State of Bombay and one in Shahdara-Delhi. Their annual capacity is estimated at 6,325 engines on one-shift basis.

(b) *Production*—Diesel engines have been produced in India for the last two decades. The importance of this industry, however, has been appreciated only after World War II which gave a fillip for higher production. Production has shown considerable increase in recent years as may be seen from the following annual production figures (Table 3).

The Prime Movers Panel in their report had recommended that during the first five-year plan of development, the target should be the production of new prime movers totalling 1.75 million k.w. capacity. The target for the oil engine prime movers is given in Table 4

TABLE 3—ANNUAL PRODUCTION OF DIESEL ENGINES IN INDIA

	Production			Total no. of engines produced	Total h.p.	Remarks
	0-10 h.p.	11-20 h.p.	Above 20 h.p.			
1947	—	—	—	685	—	Production figures h.p.-wise not available for 1947 to 1949.
1948	—	—	—	1,025	—	
1949	—	—	—	2,076	—	
1950	2,345	2,190	61	4,596	41,349	
1951	5,601	1,623	22	7,246	55,847	
Jan. 1952	557	106	—	663	3,879	

TABLE 4—TARGETS FOR OIL ENGINE PRIME MOVERS

Sizes in B.h.p.	Classification	Annual production	Total h.p.
Up to 12	F	60	30,000
12- 30	E	600	1,86,000
30- 60	D	400	36,000
60-120	C	1,000	45,000
120-500	B	2,000	42,000
500	A	5,000	60,000

### Components

In all engines manufactured in India a number of imported components are used, e.g., fuel pumps, fuel injection equipment, valves, valve springs, pistons, crank shafts, cam shafts, etc. Indigenous manufacture, however, has started for a number of components previously imported. For instance, Messrs. Kirloskar Oil Engines Ltd., used to import forgings for crank shafts, cam shafts, valve rockers, rocker lever, internal lever, connecting rods, liners and piston rings. The manufacture of these components has now been established in the country and they are now independent of the import of these items.

Similarly, Messrs. Cooper Engineering Ltd., have established the manufacture of forced feed lubricators through their sister concern Messrs. Acme Manufacturing Co., Ltd., Bombay. Messrs. Ruston & Hornsby (India) Ltd., have also become independent of imports in a number of components, such as mechanical lubricators, rubber joints, fuel tanks, splash guards, spanners, atomiser lifting plugs, oil cans, etc.

A list of imported components and accessories of diesel engines which are now being manufactured in this country is given in Table 5.

TABLE 5—COMPONENTS WHOSE MANUFACTURE HAS BEEN ESTABLISHED IN INDIA WITHIN THE LAST THREE YEARS

<i>Components</i>	<i>Manufacturers</i>
1. Forced Feed Lubricators	Acme Manufacturing Co. Ltd., Bombay
2. Splash Guards	do.
3. Crank Skew Gear Guards	do.
4. Governor Covers	do.
5. Rubber Joints	Korula Rubber Co., Bombay
6. Fuel Tanks	Fehmi Zakir Drum Manufacturing Co., Bombay
7. Splash Guards	do.
8. Air Silencers	do.
9. Fuel Tank Cocks	Annapurna Metal Works, Calcutta
10. Name Plates	New Metal Label Mfg. Co., Bombay
11. Spanners	New Standard Engineering Co. Ltd., Bombay
12. Cylinder Covers	do.
13. Governor Covers	Oriental Pressing Works, Bombay
14. Governor Gauges	do.
15. Piston Gauges	Materials from Hatimbhoy & Goolam Husein & Bros., Bombay, and made in the workshop of M/s. Ruston & Hornsby (India) Ltd.
16. Air Valve Grinding Tools	
17. Pintong with Wires	
18. Atomiser Copper Drifts	
19. Exhaust Valve Grinding Tools	Materials from New Standard Engg. Co. Ltd., Bombay and made in the workshop of M/s. Ruston & Hornsby (India) Ltd.
20. Atomiser Lifting Plugs	Hatimbhoy Goolam Husein & Bros., Bombay
21. Oil Cans	Babulal & Bros., Bombay
<i>Forgings for:</i>	
22. Crank Shaft	Premier Automobiles Ltd., Bombay
23. Connecting Rod	do.
24. Cam Shaft	Praga Tools Corporation, Secunderabad
25. Valve Rockers	do.
26. F.P. Rocker Lever	do.
27. Internal lever	do.
28. Piston Rings	India Pistons Ltd., Madras

Crank shafts and connecting rods for the Ruston engines are drop stamped from B.S.S., E.N. 6A steel.

Messrs. Cooper Engineering Ltd. manufactured during World War II some of their diesel engines with Meehanite crank shafts. These crank shafts generally proved successful,

The Indian Standard Metal Co., Ltd., Bombay, will shortly produce on experimental basis crank shafts from spheroidal graphite cast iron alloys.

In addition to pig iron and steels, non-ferrous metals, namely, "Y" metal for pistons, copper tubing for fuel connection, etc., bronzes and brasses for bushes, and aluminium are also used for the manufacture of diesel engines. In 1951 India used 409 tons of steel, 2,867 tons of pig iron and 68 tons of non-ferrous metals for manufacture of diesel engine components.

### Demand

It would be evident from the study of figures of past imports and indigenous production that the demand has been fast growing for diesel engines up to 10 h.p. This category of engines is used for driving of pumps, for irrigation purposes.

Our annual demand of engines can be roughly assessed at 55,000 engines divided into three categories: (i) 40,000 for engines up to 10 h.p., (ii) 10,000 for engines from 11 to 20 h.p. and (iii) 5,000 for engines above 20 h.p.

### Future Projects and Planning

The present capacity in the country is far lower than the assessed demand. It is of great national importance that additional capacity is set up. The development of the diesel engine industry is left to private enterprise and the Government of India have already sanctioned six new projects which, when fully realised, would add to the existing capacity (Table 6).

TABLE 6—PROPOSED NEW INSTALLED CAPACITY

0-10 h.p.	11-20 h.p.	Over 20 h.p.	Total
14,700	3,500	400	18,600

All the sanctioned schemes envisage manufacture of engines of known makes, namely, Deutz, Diahatsu, Russel Newberry, Imperial Kieghley, Yanmar and Mitsubishi. The name of the engine manufactured to these designs in India will be 'Central'.

# Some Aspects of Improving the Performance of Automobile Engines Running on Producer Gas

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It is of vital interest to our country with meagre petroleum resources to utilise producer gas derived from vegetable matter and low grade coal, which are available in plenty, as partial substitute for petrol. The main drawback, inherent to the use of producer gas in petrol engines, is loss of power. To counteract this, the technique of using producer gas as fuel in I.C. engines can be considerably improved by introducing air-fuel mixtures into the cylinder at high pressures, in other words, by supercharging. There has always been a controversy about supercharging, but from the point of view of economy, an exhaust turbo-supercharger is to be preferred.

In an automobile engine the load factor depends on the speed of the vehicle, the road which the vehicle is negotiating, the weight carried, etc. Accordingly, the energy available in the exhaust gases will continuously vary and this in turn will affect the running condition of the supercharger. Hence throughout the operating range of the vehicle the complete benefit from the supercharger characterised by the design point will not be available.

At low load factors the necessity for a supercharger is for all practical purposes not pronounced. An exhaust turbo-supercharger, under these conditions, may just overcome the pressure drop in the producer plant. At higher loads the energy in the exhaust gases increases and so does the power output on account of supercharging.

There are two alternatives in the use of the supercharger. A mixture of air and producer gas can be sucked by the supercharger and supplied to the engine under increased pressure. In this case the engine is supercharged while the gas plant works under atmospheric pressure. The mass flow in the producer plant increases due to increased pressure difference due to the blower suction.

On the other hand, the supercharger can supply air both to the engine and to the producer gas plant for generating gas under pressure.

The increase in power is due to the cylinder drawing in a bigger quantity of fuel-air mixture at each piston-stroke and also due to the work during the suction operation being eliminated which is otherwise considerable.

Supercharging has another special advantage: Engines originally built for petrol, for example, need not be changed over to a higher geometric com-

pression ratio because the supercharging process itself generates the necessary higher pressure required.

In fact, it is seen from the test results (Fig. 1) on an engine run by wood gas at compression ratios 4.5 and 7.5 that the power increase due to supercharging is much more considerable than that due to the increase in compression ratio.

It is also clear from Fig. 1 that for an increase of compression ratio from 4.5 to 7.5 the power increase is about 10 per cent whereas the power increase due to supercharging is nearly 50 per cent. Another significant fact is that when the compression ratio is increased from 4.5 to 7.5 and then supercharged the gain in power as compared to low compression ratio supercharged engine is negligible. That is, the increase of compression ratio involving changes in the engine is unnecessary when supercharging is used.

In the present series of experiments a positive blower driven electrically has been used for preliminary investigations. The object of using a variable speed positive blower is firstly to determine the useful boost pressure with which the performance of the engine, with gas, would reach the performance when run on petrol ; secondly to study the behaviour of the gas plant through which the air is sucked and thirdly to find the power consumed by the blower.

### Test Set Up and Procedure

The engine used for the experiments is a reconditioned six-cylinder petrol engine (Fig. 2) rated at 30 h.p. An orthodox cross draught mobile gas plant with normal coolers and filters is installed as a stationary unit. The supercharger is a three-lobed Roots blower separately driven by a variable speed d.c. motor.

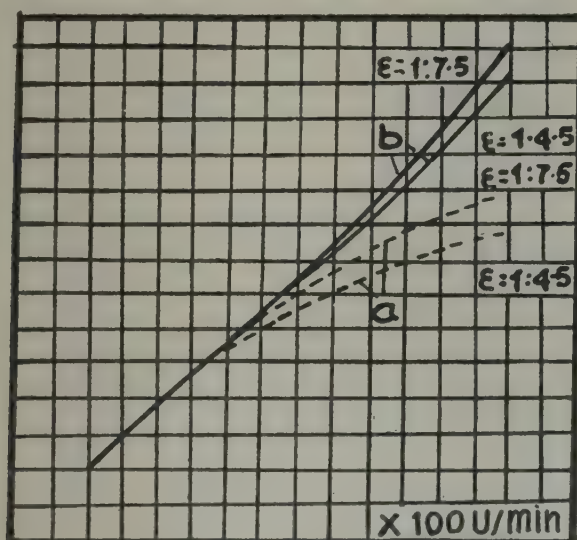


FIG. 1—POWER INCREASE OF A PETROL ENGINE WITH UNCHANGED COMPRESSION RATIO 1: 4.5 AND OF A PETROL ENGINE WITH INCREASED COMPRESSION RATIO 1: 7.5 AS A FUNCTION OF THE SPEED

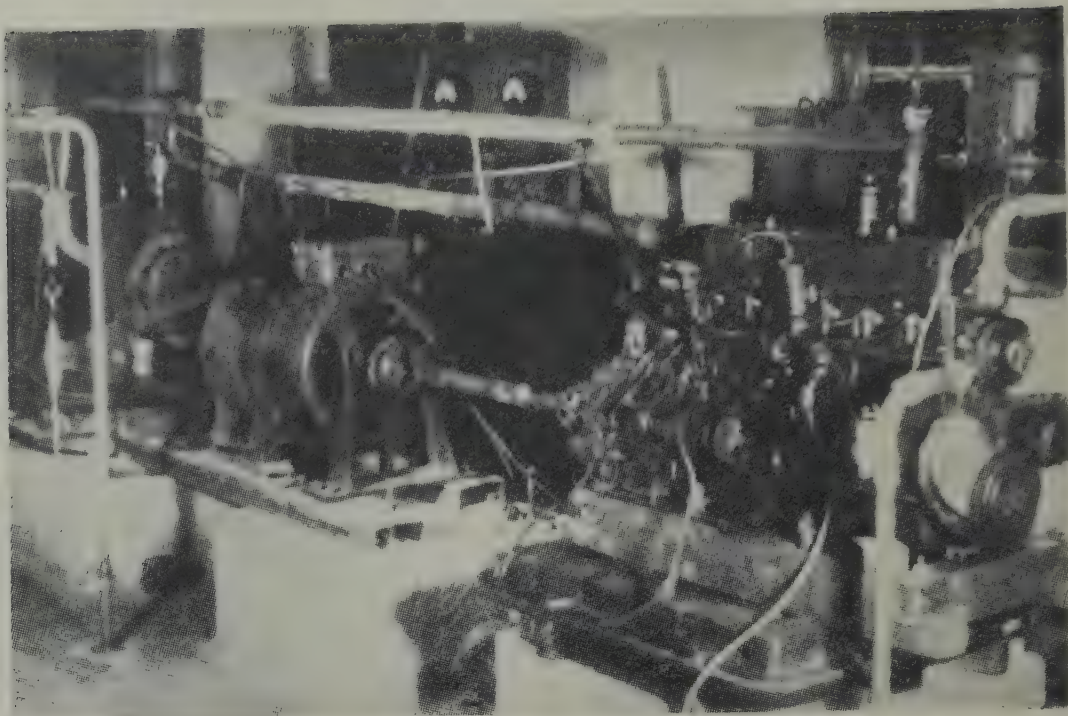


FIG. 2—TEST SET-UP

Gas and air are sucked through a gas carburetor so that the fuel-air mixture strength could be controlled. The delivery pressure is recorded by a low range pressure gauge and this pressure can be regulated by means of a by-pass valve.

The engine characteristics on petrol are first recorded and the gas plant is connected to the engine. The d.c. motor consumption gives approximately the power absorbed by the blower.

### Results

The results obtained with supercharging on the test bed are reproduced in Fig. 3. For purposes of comparison the torque developed by the engine when running on petrol is given and the considerable gain due to supercharging is obvious ; nearly 80 per cent of the power on petrol is available. Of course, from this the power required to run the blower is to be deducted.

The results recorded on the test bed will generally be valid for practical service. The same vehicle under identical conditions can negotiate a grade in higher gear when supercharged than without supercharging. As the engine does not draw in fuel and air directly but gets it pumped by a blower, which is to be brought to speed, it is better to change the gear a little earlier. The technique of driving is subjected to a slight modification.

### Conclusion

From the preliminary set up and the running of the engine the following conclusions are drawn :

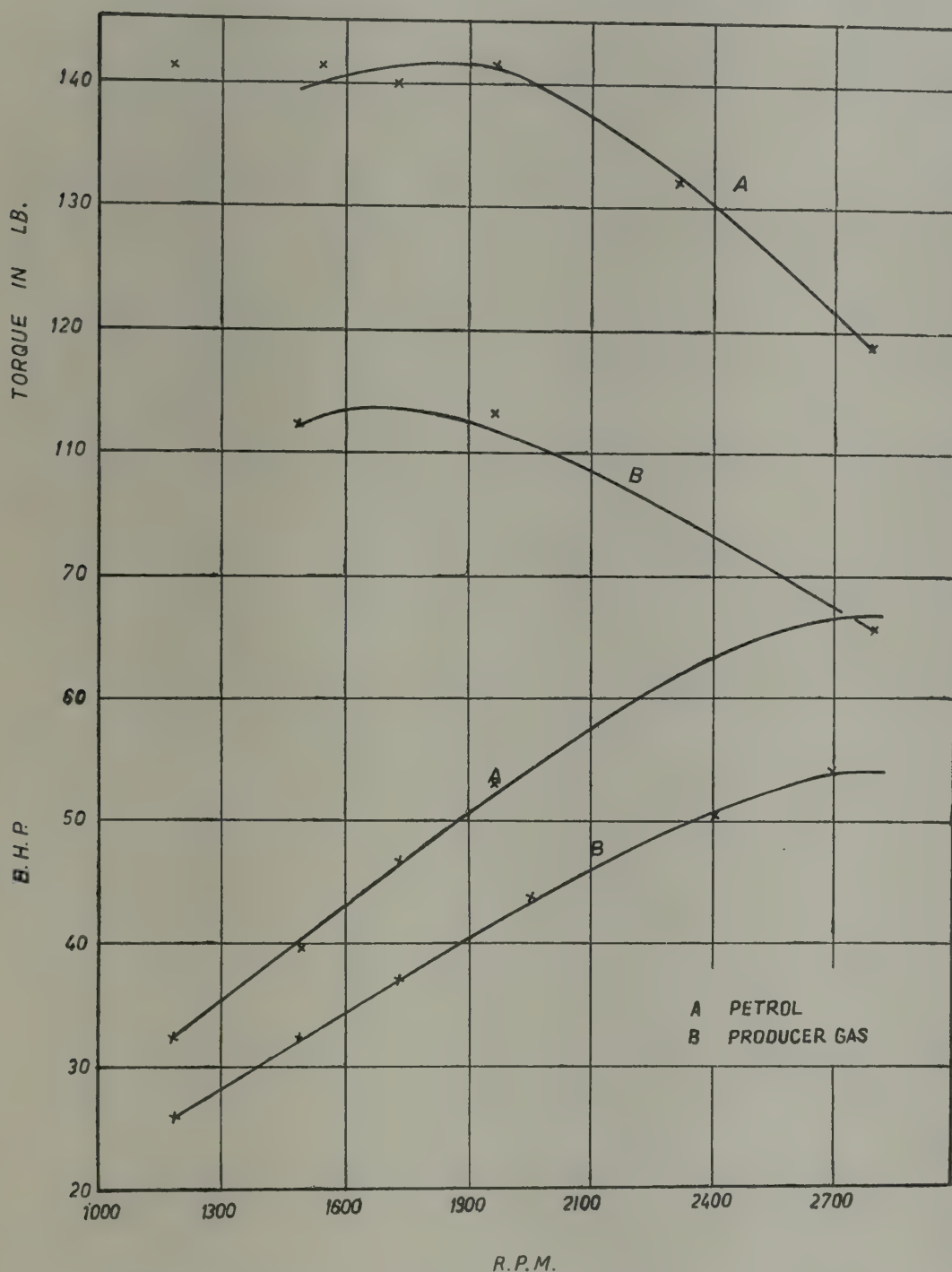


FIG. 3—EFFECT OF SUPERCHARGING ENGINES RUN ON PRODUCER GAS

Since in the case of a positive blower the mass flow is a function of speed, and it takes time to build up the pressure the correlation of engine speed and blower speed gives considerable difficulty.

At low speeds the blower actually behaves like a valve on account of the rotors which seal the passage and as such idling and low speed running of the engine is difficult.

These difficulties will not be pronounced in the case of centrifugal blowers where the pressure is a function of speed. The engine is always supplied with its requirement at the desired pressure. Since the speed of the centrifugal blower should be high it would be preferable to drive it by an exhaust gas driven turbine. The exhaust gas turbo-supercharger can easily be located wherever required.

When air and gas are both sucked it is found difficult to get the correct air-fuel ratio and it is also found that the gas plant is a little sluggish in responding to engine requirements. It is preferable to pressurise the gas plant. This requires very little modification to the gas plant. In such a system it would be advantageous to locate the turbo-blower in front of the gas plant. This will be very simple in the case of exhaust driven turbo-blowers as it requires only a little extension to the exhaust connections.

Experiments in this direction using a turbo-supercharger and pressurising the gas plant may give better results, and the modified set up can be put on road for further tests.

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# A New Hot Air Engine

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## Introduction

This paper describes a new hot air engine that is being developed in the Department of Internal Combustion Engineering, Indian Institute of Science, Bangalore, under a grant from the Council of Scientific and Industrial Research, New Delhi.

The hot air engine aims at assisting in the development of irrigation, agriculture, cottage industries and allied fields in our country by its ability to make use of easily available local fuels to produce cheap power with little demand on high accuracy production and on skilled maintenance and operation.

Fig. 1 shows, in diagrammatical representation, the simplest form of the cycle used in the engine. Atmospheric air, drawn in through the inlet A, is compressed in a compressor B, heated in a heat exchanger C and is then allowed to expand in an expansion motor D. The clean hot air from the expansion motor is used to burn fuel in a combustion chamber E and the hot gases are sent into the hot side of the heat exchanger C to heat the incoming compressed air. From the heat exchanger C, the hot gases are allowed to escape to the atmosphere through the exhaust F. These exhaust gases or, alternatively, the hot air from the expansion motor, may be used for a variety

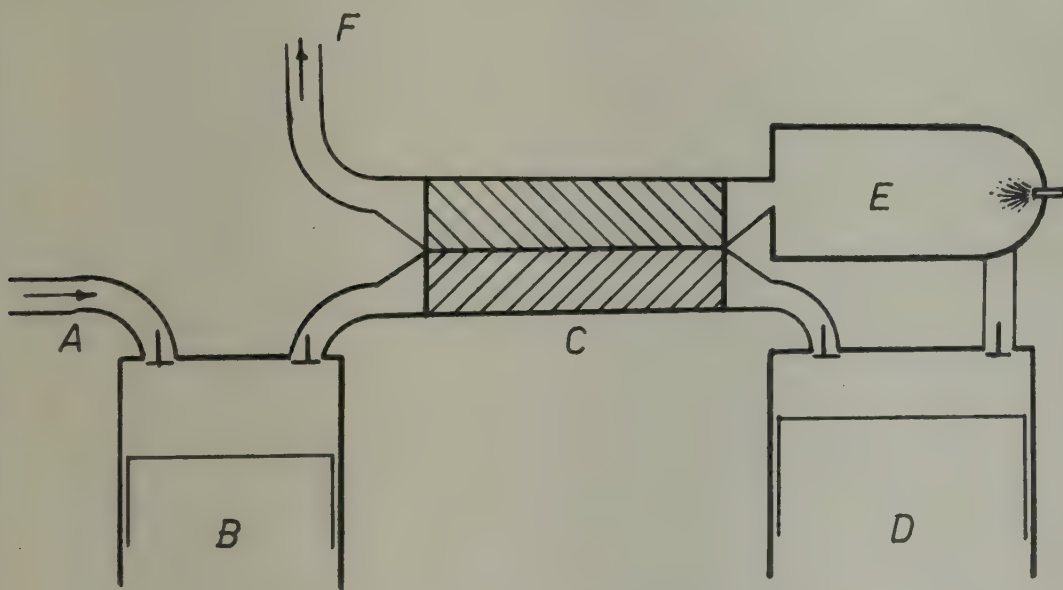


FIG. 1.—THE SIMPLE HOT AIR ENGINE CYCLE

- |                   |                       |
|-------------------|-----------------------|
| A. Air inlet      | D. Expansion motor    |
| B. Compressor     | E. Combustion chamber |
| C. Heat exchanger | F. Exhaust            |

of purposes such as drying the fuel. The important possibility of using a single cylinder for compression and expansion is discussed later in the paper.

It will be noticed that the residues of combustion do not come into contact with any of the moving parts of the engine at any time during the process, thus introducing the possibility of making use of an inferior grade of fuel such as peat or lignite.

Within the last three to four years, this cycle as well as some variations of it, have been considered or tried for high-output gas turbines. Reference to this work may be found in the bibliography at the end of this paper. The application of the cycle to low-output reciprocating units, however, has not been attempted by any one, as far as it could be ascertained. This application introduces fresh problems and requires new techniques for their solution.

### Theory

The air standard efficiency of the cycle is given by  $\eta = 1 - (E)^{\frac{1}{k-1}}$  when  $\eta$  is the efficiency,  $E$  the compression ratio and  $k$  the adiabatic constant.

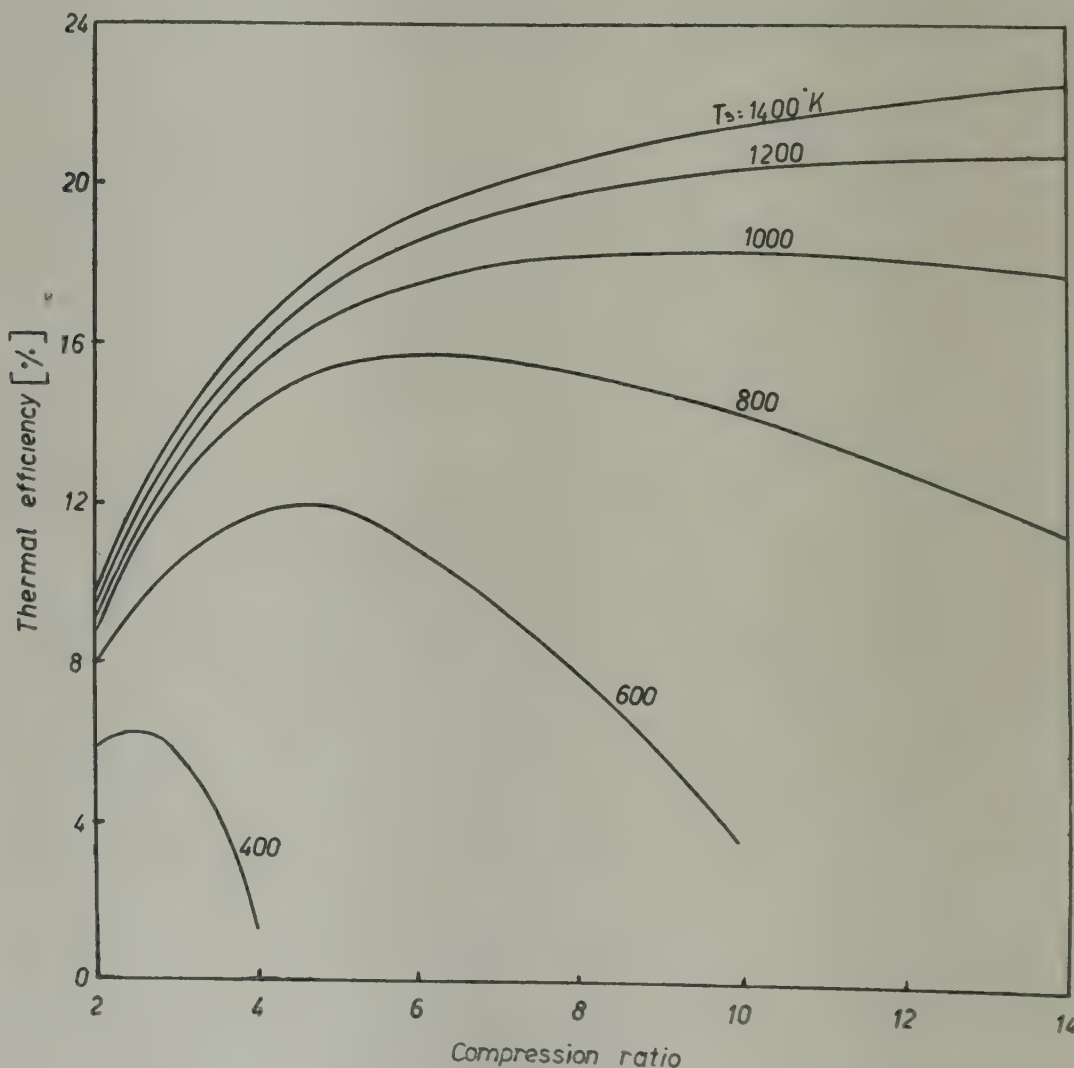


FIG. 2—THERMAL EFFICIENCY OF THE SIMPLE HOT AIR ENGINE CYCLE

The actual thermal efficiency and specific output of the engine have been calculated for average conditions for simple compression and expansion plotted in Figs. 2 and 3 respectively. Both the thermal efficiency and specific output curves are plotted against compression ratio with  $T_3$ , the temperature at the beginning of expansion, as the parameter. These curves show that while the specific output increases almost directly in proportion to  $T_3$ , the thermal efficiency does not increase quite so rapidly.

Though the thermal efficiency is low as compared to the usual internal combustion engine, this is more than offset by the extremely low cost of the inferior fuel that can be used. The following table shows the relative fuel costs per unit of power output with reference to the use of low grade coal in a hot air engine assumed to be unity.

Type of Engine	Type of Fuel	Relative Cost
Hot Air Engine	Low-grade coal	1.0
Hot Air Engine	Pulverised coal	1.3
Diesel Engine	H.S.D. Oil	3.7
Petrol Engine	petrol	10.1

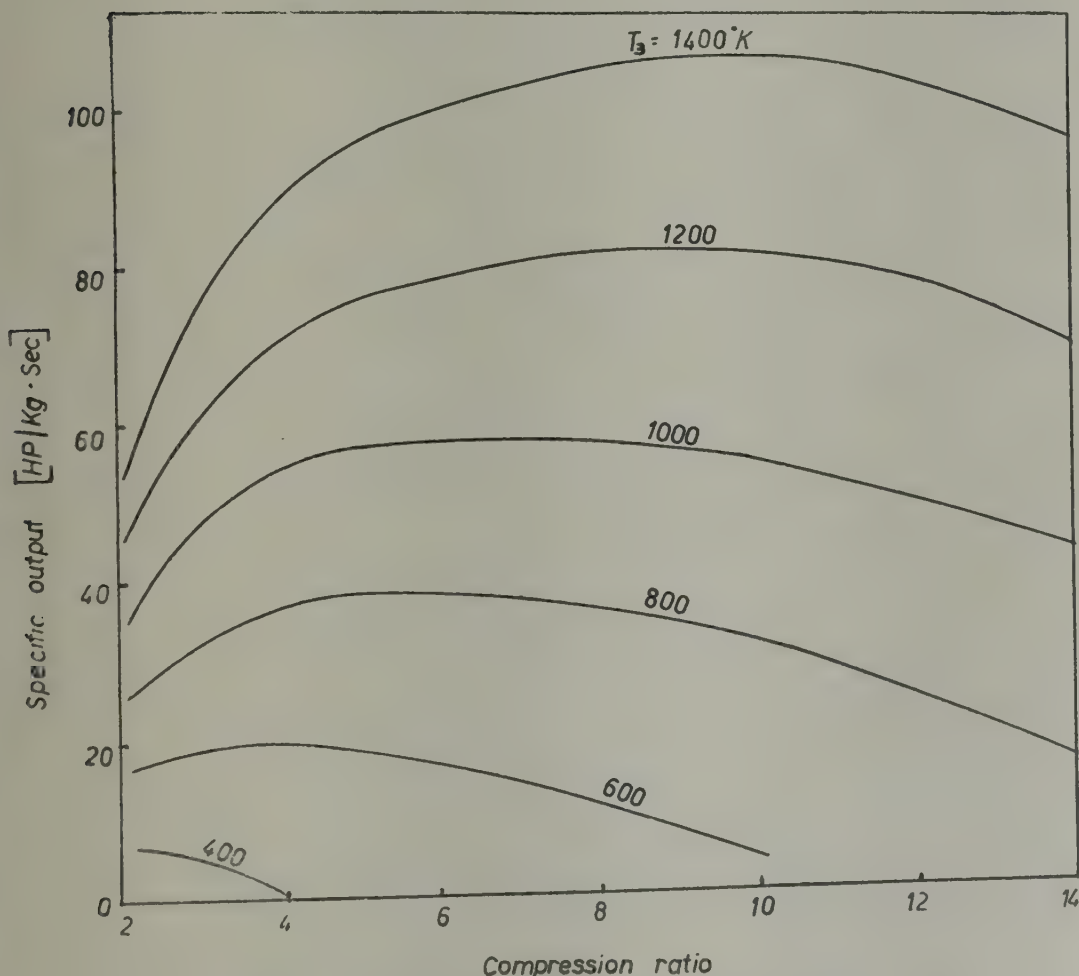


FIG. 3—SPECIFIC OUTPUT OF THE SIMPLE HOT AIR ENGINE CYCLE

It is also noteworthy that the specific output (h.p./kg. sec.) compares favourably with any other type of cycle.

One method of improving the thermal efficiency is to make use of multi-stage compression and expansion. Fig. 4 shows the efficiency and Fig. 5, the specific output for a two-stage process. These curves show that an optimum range of values exists both for the compression ratio and the temperature at the beginning of expansion,  $T_3$ . If the compression ratio is increased beyond certain limits, the specific output falls, while if  $T_3$  is increased the thermal efficiency either falls or does not improve appreciably. Similar considerations apply also to the three-stage process.

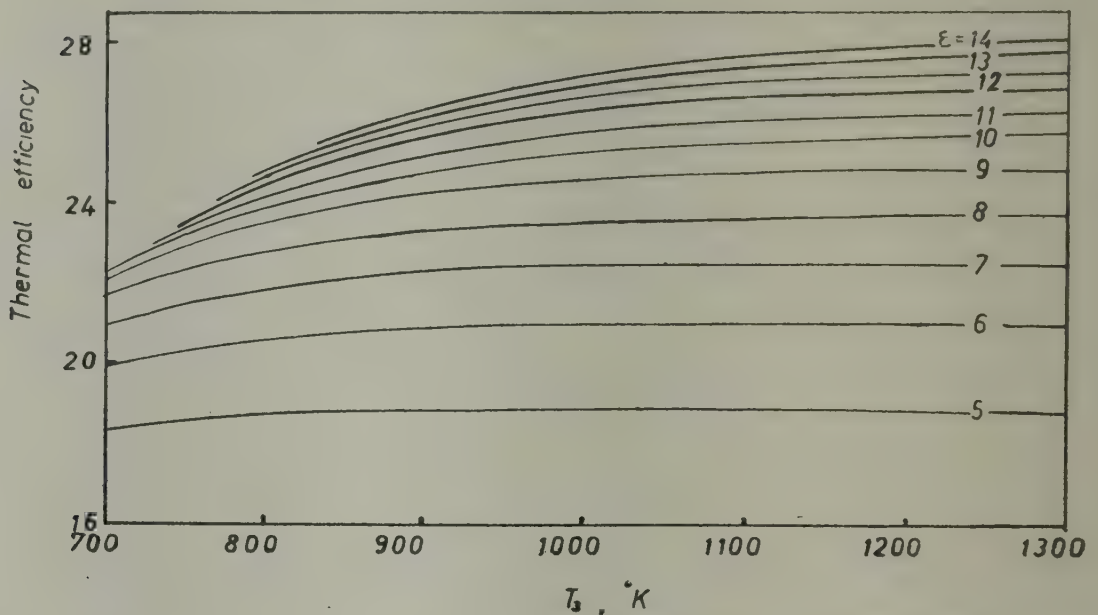


FIG. 4—THERMAL EFFICIENCY FOR TWO STAGE PROCESS WITHOUT REHEAT

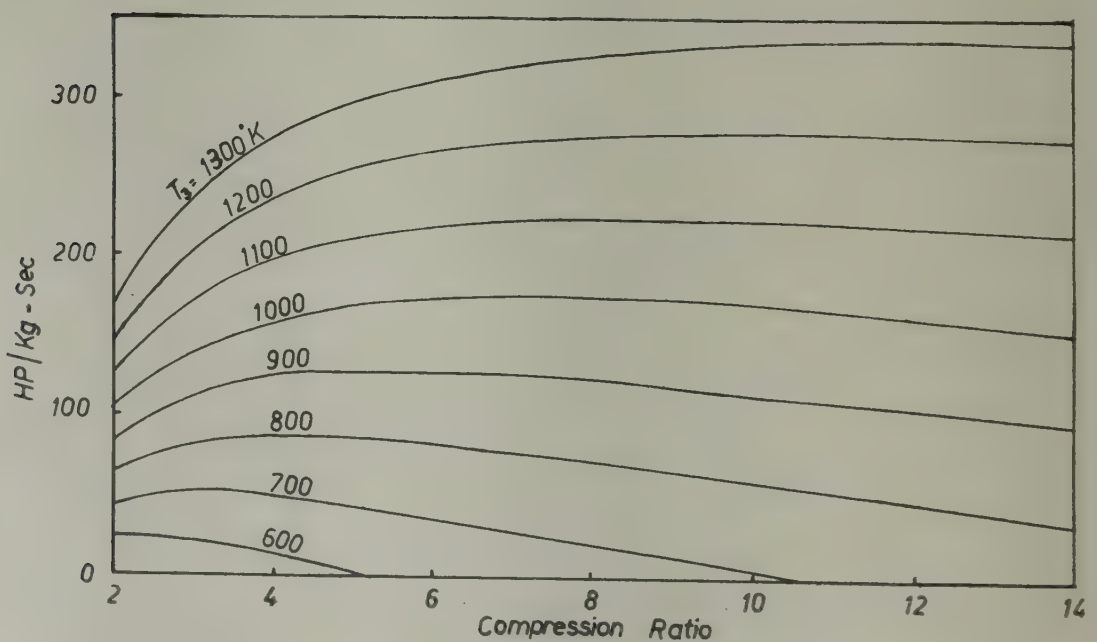


FIG. 5—SPECIFIC OUTPUT FOR TWO STAGE PROCESS WITHOUT REHEAT

The positions of the maxima of the specific output and thermal efficiency curves are plotted in Fig. 6. These curves show that the single-stage process is preferable to the others as the specific output and thermal efficiency curves are nearer to each other than in the case of the two- and three-stage processes and therefore any suitably chosen design value for compression ratio and maximum temperature would give both maximum specific output and maximum thermal efficiency. Though it is not possible to design simultaneously for both maximum thermal efficiency and maximum specific output in the case of multi-stage units, this fact might be ignored as the actual increase in both efficiency and specific output is considerable as compared to single-stage units. The numerical value of the limits are altered by the amount of reheat applied between stages during expansion and amount of recooling applied between stages during compression. Reheat also increases the thermal efficiency and specific output as shown by the full and dotted lines respectively in Fig. 7.

Various other methods can also be used to increase the efficiency. For example, an exhaust gas turbine might be incorporated at the end of the heat exchanger or the expansion motor to drive either the compressor or another primary compressor or a rotary type of heat exchanger.

### Design

The compressor and expansion motor do not present any important problems except for the provision of a method for the variation of the valve timing and hence the cut-off ratio during operation in order to control the power output. This might be in the form of a D-slide valve as in a steam engine or a variable lift cam whose lift could be made to vary with axial movement of the camshaft.

The combustion chamber may be designed to make use of any type of fuel, solid, liquid or gaseous, depending upon local conditions. If pulverised fuel were to be used, a cyclone combustion chamber with spirallie heat exchanger tubes in parallel or counter-flow arrangement inside the chamber might be designed. For other types of fuels, conventional furnaces can be used.

The design of the heat exchanger presents more complex problems as it is not easy to evaluate all the unknown variables involved with any degree of simplicity or certainty, or by means of theory alone. The calculation of heat transfer rate is complicated by the presence of oscillations of large amplitudes and the consequent production of unknown pressure gradients. The effect of the relative valve timing between compressor and expansion motor further complicates matters. The pressure fluctuations due to the compressor as well as the oscillations imposed by the valves and ducts would probably increase the turbulent characteristics of flow and consequently the rate of heat transfer.

The effect of the pressure fluctuations on the pressure drop in the heat exchanger is another unknown factor. The pressure drop due to uniform flow of gases (either laminar or turbulent) during heat transfer, together with the pressure drop due to bends, ducts, constrictions, etc., can be calculated

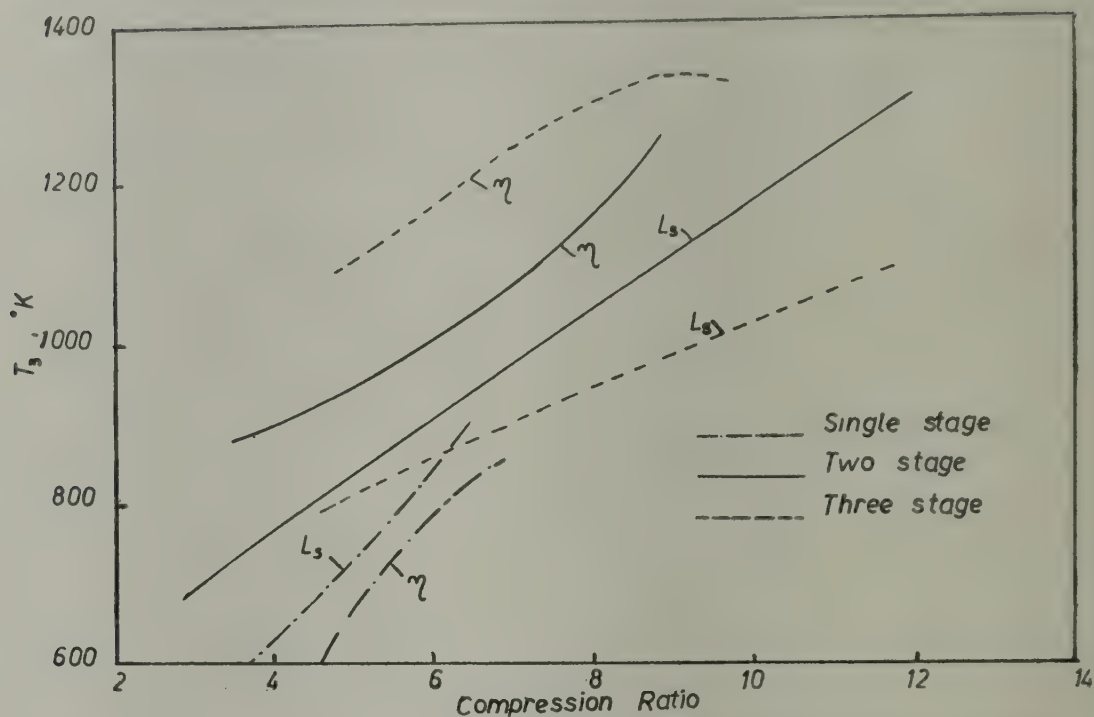


FIG. 6—LINES OF MAXIMUM THERMAL EFFICIENCY AND MAXIMUM SPECIFIC OUTPUT

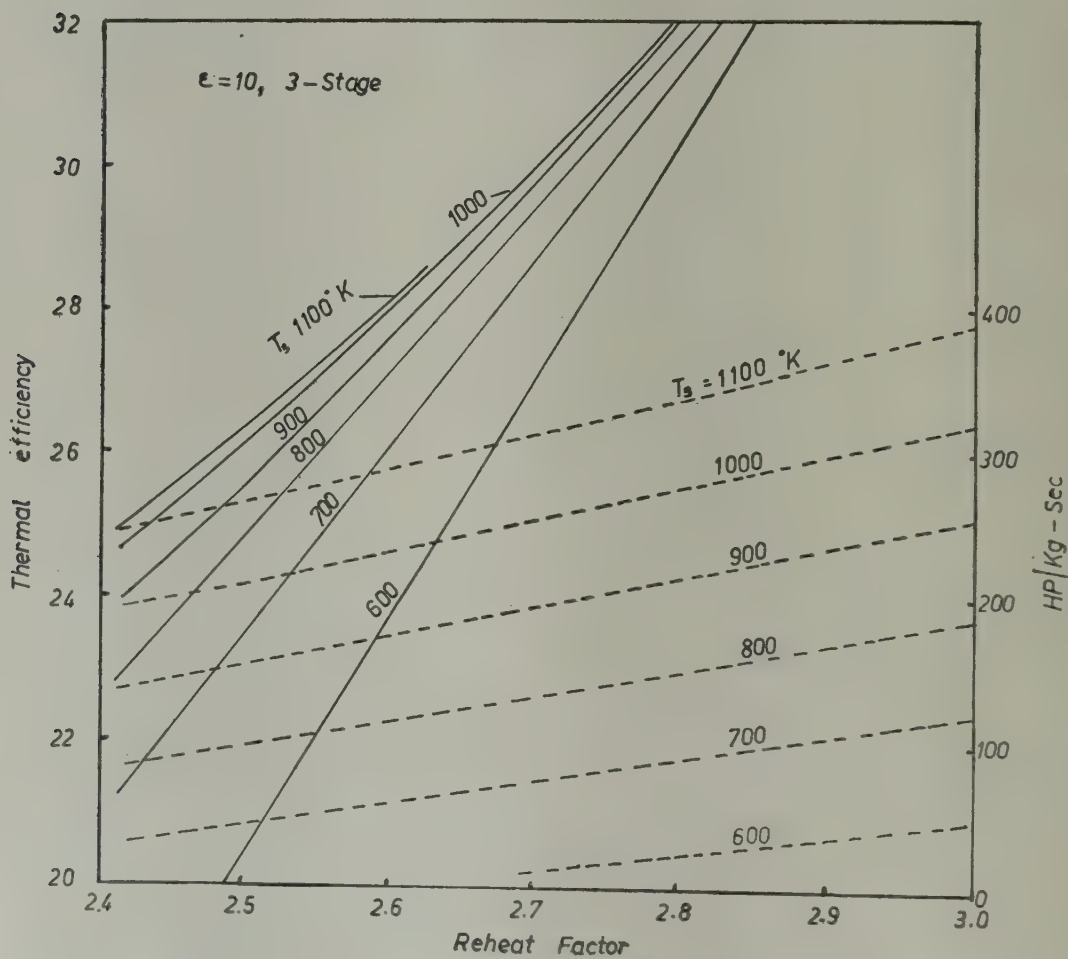


FIG. 7. EFFECT OF REHEAT FACTOR ON THERMAL EFFICIENCY AND SPECIFIC OUTPUT

with reasonable accuracy by conventional methods. The best exchanger should be designed for the lowest pressure drop as otherwise the pumping horsepower would increase and the thermal efficiency would fall. When pressure fluctuations are superimposed on the flow, the efficiency of the unit may be improved by judicious use of the fluctuations as in the case of a two-stroke engine where proper design of the exhaust pipe, results in as much as 30 per cent more power output.

Many of these problems may be either solved or completely eliminated by the use of rotary heat exchangers, especially those in which heat is transferred by making use of fluid at or near the critical state.

The cut-off ratio plays an important role in the inter-relations of the different factors and is defined as the ratio of the actual volume of hot air admitted into the expansion cylinder to the total volume of the cylinder and is the inverse of the expansion ratio. It increases with the temperature increase in the heat exchanger, and with the ratio of expansion cylinder volume to compressor cylinder volume but decreases with compression ratio. For any particular unit, any regulation of heat exchanger must be accompanied by a suitable alteration of the cut-off ratio.

Another problem in the successful design of an actual power plant is that of its layout. One method of simplifying the layout as well as reducing considerably the capital cost and size of power plant is the use of a single cylinder for both compression and expansion. In this modification, which may be called the 'four-stroke' version, the first two strokes are used for suction and compression of atmospheric air while the third and fourth strokes are used for the expansion and exhaust of the hot air from the heat exchanger. Though it is not ideal for the compressor and expansion motor cylinder volumes to be equal, yet it is possible to design this single-cylinder unit also to have a reasonable efficiency by suitably arranging the cut-off ratios and valve timing. The unit may be used by itself or may be used just for the necessary expansion while useful expansion may be performed in a separate cylinder (necessary expansion denotes the expansion necessary to overcome all the losses in the engine, and useful expansion, the expansion necessary to produce useful power). A number of these units may also be attached to a common furnace and a heat exchanger.

The first prototype built in this Department is by converting a single-cylinder diesel engine into a compressor and a twin-cylinder petrol engine into an expansion motor by alteration of their camshafts and timing gear and therefore their valve timings. For the sake of flexibility during initial experiments, the compressor is run by a separate motor while the expansion motor is coupled to an electrical dynamometer to measure the expansion power output. A lignite-fired furnace has been constructed from a mild steel shell lined inside with insulation and fire-clay bricks and is fed by a double hopper. The furnace gases, cooled by secondary air from a separate blower, are led through a long cast-iron pipe in which are placed some locomotive superheater tubes to form a combined parallel and counterflow tubular heat exchanger with hot gases outside and compressed air inside the tubes. This prototype

has been built to obtain some preliminary data quickly and cheaply so that a complete power plant could be designed later with greater certainty regarding some of the factors mentioned above.

### Acknowledgments

This investigation, initiated by Prof. H. A. Havemann, in the Department of Internal Combustion Engineering, Indian Institute of Science, Bangalore, is being carried out under the auspices of the Council of Scientific and Industrial Research, New Delhi. The author wishes to express his appreciation and thanks to both these bodies for the facilities which have been placed at his disposal.

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# Low Pressure Fuel Injection Systems for Diesel Engines

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The efficient working of a high pressure injection system requires the use of the very best materials available and an extremely high degree of accuracy and finish in manufacture. In the case of very small pumps the technical difficulties in manufacture tend to become almost insurmountable and the cost of production becomes excessive, so that the injection equipment forms a major part of the total cost of an engine.

An efficient and reliable low pressure injection unit would not require precision parts and would be simple to maintain in good order. It would be less expensive both in manufacture and in use. A diesel engine with low pressure injection of fuel would undoubtedly possess a strong attraction for the diesel operator.

The low pressure systems tried in the past restricted themselves to elimination of the high pressures only in that part of the system external to the engine cylinder. That is to say, while the fuel pump and connecting lines were under low pressure the injector itself was fitted with a device to force the fuel under high pressure through the atomiser jets. In this connection, the Cummin's system and the General Motors' Unit Injector may be cited as examples. These systems, while they relieve the pump from the severe operating conditions, do not really offer a solution to the problem.

A system which aims to eliminate the high pressure pump altogether has been achieved by Dr. L'Orange in Germany. In this system a narrow venturi is interposed between an auxiliary combustion chamber and the main cylinder space (Fig. 1). During the compression stroke of the engine, a large pressure difference is created across the throat because of the high resistance to flow offered by the constriction. This differential head is utilised to spray fuel into the ante-chamber through fine capillaries communicating on either side of the throat. Combustion is thus started in the ante-chamber and takes place at approximately constant volume. The resultant pressure rise reverses the pressure head and the fuel remaining in the connecting passages is blown into the main cylinder space. The rest of the combustion then takes place at approximately constant pressure.

Dr. L'Orange used this principle successfully in both two- and four-stroke cycle engines of capacities ranging from 100 c.c. to 2,000 c.c. and speeds of 2,400-6,000 r.p.m. Admission of fuel to the capillary passages was controlled by a spring-loaded automatic non-return valve in the case of two-stroke engines.

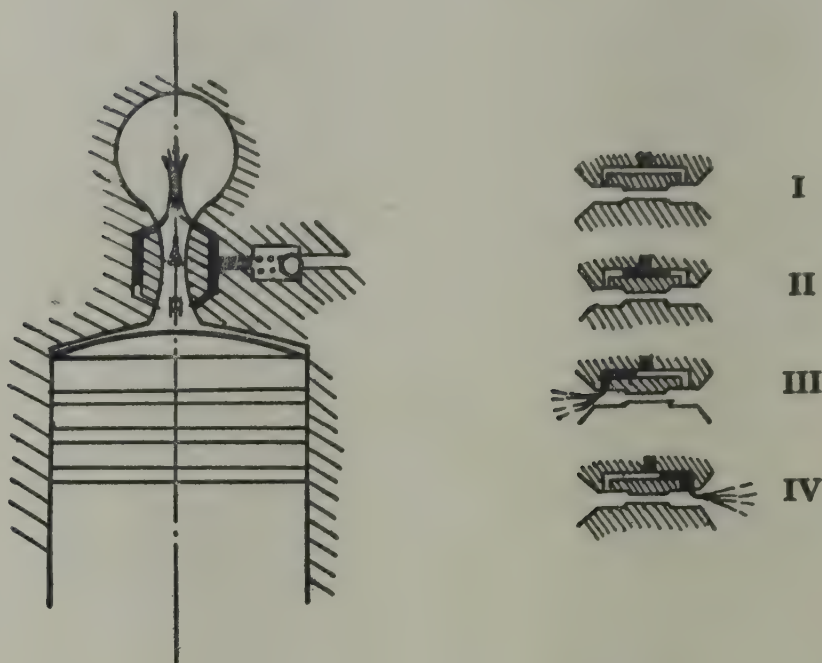


FIG. 1—L'ORANGE PUMPLESS INJECTION SYSTEM. I. After Combustion ; II. After Induction or Scavenging ; III. Injection into Chamber ; IV. Injection into Cylinder Space

Mechanically operated valves were used on the four-stroke engines. Specific power output and fuel consumption were claimed to be comparable to that of normal diesel engines.

On a similar principle, a low pressure fuel injection system is being developed in the Internal Combustion Engineering Department of the Indian Institute of Science. A preliminary theoretical study showed that the pressure differential is closely inter-related to the engine speed and the size of venturi-throat and to a smaller extent to the relative proportions of the ante-chamber and main cylinder. Also, in any given case, there would be one particular speed or at best a narrow range of speed, at which the best operating conditions for the engine would be attained.

For experimental work a small two-stroke diesel engine has been chosen on account of the simpler fuel control system and the simplicity of the engine itself. The test unit is a 4 in.  $\times$  4 in. engine with crankcase scavenge, developing 3 h.p. at 600 r.p.m.

A new cylinder head has been designed for this engine so as to incorporate the requisite features for operating on the principle of differential-pressure-injection of fuel. The modified cylinder head is shown in Fig. 2. It is made in two parts in order to simplify the casting, the water jacket being separate from the cylinder head proper. Provision has been made to study the effect of varying the size of the ante-chamber (1) and the venturi-throat (2) by making them as separate parts which can be easily inter-changed with similar pieces.

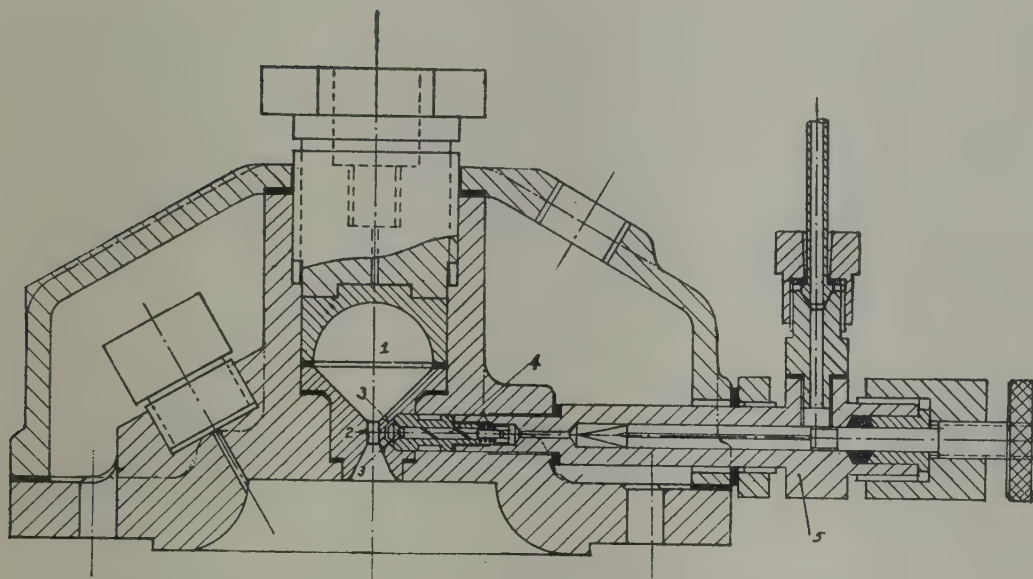


FIG. 2—CROSS-SECTION OF CYLINDER-HEAD FOR EXPERIMENTAL TWO-STROKE DIESEL ENGINE WITH LOW-PRESSURE INJECTION

The top half of the ante-chamber has been left uncooled so as to provide a hot surface against which the incoming fuel spray will impinge. The capillary passages for fuel are included in the part which forms the venturi. Fuel admission is controlled by a spring-loaded non-return valve (4) and a metering unit (5). The metering unit consists of a jet and a movable needle. The compression ratio has been raised slightly above the original value.

Considerable difficulty was experienced in the initial stages in making the non-return valve perfectly leak-proof and function properly over a prolonged period. Different materials and various designs of valves were tried and with slight modifications in valve design and disposition this problem has been partially solved. It was also found that the valve would operate more satisfactorily with a slight positive pressure feed of the fuel, instead of relying only on gravity flow. A plunger pump was used to pressurise the system.

Fig. 3 shows the test set-up.

Several short runs of the engine have been made so far but no quantitative results can be presented at this stage. However, it can be stated that the engine starts readily without noticeable knock and runs smoothly, though the exhaust is somewhat smoky. It has also been noticed that some unburnt fuel is present in the exhaust. This indicates that fuel consumption would be likely to be on the high side. Efforts will be directed in the first instance towards obtaining good running conditions rather than low fuel consumption. With further design changes, now being carried out on the metering system and the flow passages, greater economy in fuel is expected.

The favourable results so far obtained point to the possibility of wider application of this method, in future, on engines of small output. High rotational speeds should not present any difficulties, since the system can be proportioned to suit the speed. The small medium speed two-stroke engine

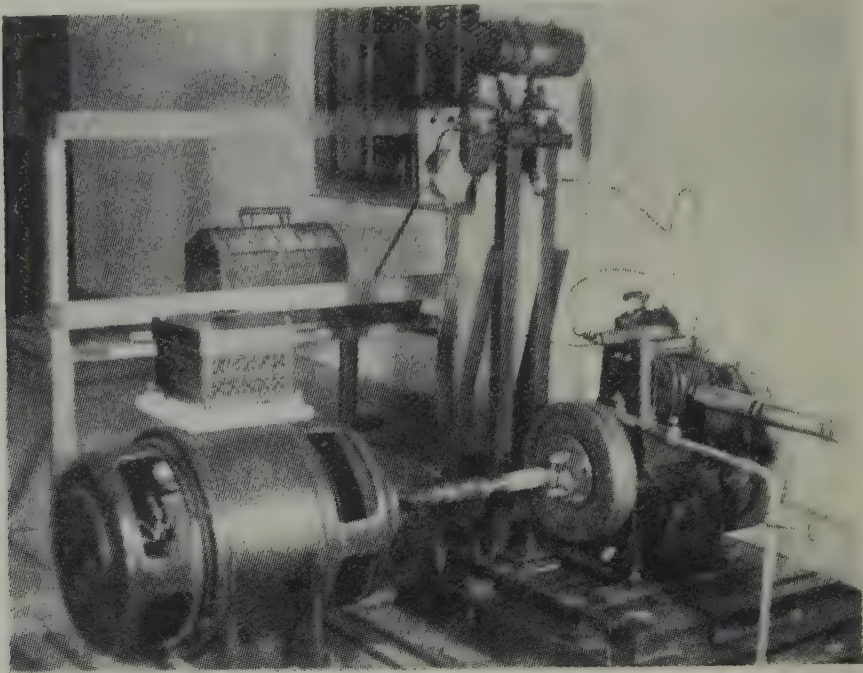


FIG. 3—TEST SET-UP OF L.P. FUEL INJECTION SYSTEM FOR DIESEL ENGINES

with crankcase scavenging and low pressure fuel injection would be a highly attractive unit on account of its simplicity and low cost.

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# Scavenging of two Stroke Diesel Engines With Particular Reference to the Design of a U-Type Experimental Unit

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The endeavours of an engine designer are always directed towards obtaining a given output with the minimum expenditure of material. The stroke volume of an engine forms an useful measure of the expenditure of material. The utility of the engine is higher, the higher the specific output, *i.e.*, power per unit stroke volume of the engine.

The specific output of an engine is independent of its dimensions and is a function of the brake mean effective pressure (B.M.E.P.) and the speed of the engine. In two-stroke engines, the B.M.E.P. depends mainly on the scavenging efficiency, *i.e.* the efficiency with which the cylinder is filled with fresh charge, and the degree of supercharge. Uniflow scavenged two-stroke diesel engines, which have a high scavenging efficiency and provision for supercharge, appear to be the natural consequence of the efforts of the engine designer.

Among the successfully developed uniflow scavenged two-stroke diesel engines are the Junker's opposed piston engine and the General Motors engine with ports for inlet air and poppet valves for exhaust. It can be said that no attention has been paid so far to develop the U-type version of the uniflow scavenged engine.

This paper briefly describes the design features of a U-type experimental unit, designed in the department of I.C.E., on the basis of certain theoretical investigations carried out to determine the possibilities of such an engine.

## Scavenging Systems

There are three main systems of scavenging a two-stroke engine, namely, cross scavenging, loop scavenging and uniflow scavenging. Uniflow scavenging gives the maximum scavenging efficiency and so the B.M.E.P. is maximum in this system. Opposed piston engines and engines with piston controlled ports for inlet air and poppet valves for exhaust or vice versa, work on this principle. U-type two-stroke diesel engines also work on the same principle.

## U-Type Two-Stroke Diesel Engines

U-type two-stroke diesel engines have a common combustion chamber for two parallel cylinders. In one of the cylinders which may be called the

inlet cylinder, inlet ports are situated while in the other, the exhaust cylinder, are exhaust ports. The inlet and exhaust ports are controlled by suitably phased pistons operated by the same crankshaft.

The simplest possible arrangement for operating the two pistons is a double throw crankshaft with suitably phased cranks and separate connecting rods. It is also possible to obtain the necessary phase difference between the pistons by means of either an articulating connecting rod system or a forked type connecting rod actuated by a single throw crankshaft. The present investigation covers only the case of the double throw crankshaft.

### Theoretical Investigation

At the end of the power stroke of the engine, the exhaust ports are uncovered by the exhaust piston which has a lead of a few degrees over the inlet piston. The cylinder pressure is brought down to atmospheric pressure by the discharge of combustion products through the exhaust ports. Now scavenge air enters the inlet cylinder through the inlet ports uncovered by the inlet piston, scavenges it and passes into the exhaust cylinder through the throat formed by the combustion chamber. After the exhaust cylinder is scavenged, the inlet and exhaust ports are closed simultaneously, in the case where there is no provision for supercharge. When there is provision for supercharge, the inlet ports are closed later than the exhaust ports.

So in U-type engines, while calculating the scavenge pressure, it is necessary to take into account the pressure drop caused by the throat, in addition to that caused by inlet and exhaust ports. The throat area depends mainly upon the shape and volume available for shaping the combustion chamber which in turn depends on the compression ratio. The throat area is very small in diesel engines because the compression ratio usually ranges between 14:1—18:1, depending upon the dimensions of the engine. Hence it is necessary to investigate the effect of this throat on the scavenge pressure, in order to estimate the possibilities of such engines.

In a specific example, the effect of the throat on the scavenging pressure has been estimated for the following cases: (1) with crankcase scavenging and (2) blower scavenging with no provision for supercharge.

### Crankcase Scavenging

The pressure drop caused by the throat has been estimated for a rotational speed of 600 r.p.m. and found to be not very appreciable. By slightly increasing the port dimensions, the effect of this pressure drop can be counteracted and the same quantity of air that would flow through the system if there was no pressure drop can be made to flow through the engine.

The pressure drop increases with increase in speed. Since the pressure that can be attained in a crankcase pump is fixed by the dimensions of the engine and cannot be easily increased, there is a limiting speed, fixed mainly by the special features of the U-type engine, beyond which there is an appreciable reduction in output. For this case the limit appears to be well above that

determined by other considerations like lubrication of piston, piston-pin, etc.

The B.M.E.P. has been calculated for a crankcase scavenged engine having a displacement volume of 93.16 cu.in. and working at 600 r.p.m. in the three following cases: (1) Cross scavenging, (2) Loop scavenging and (3) U-type uniflow scavenging. This calculation shows that the U-type engine gives 52 per cent more output than a cross scavenged engine and 31 per cent more output than a loop scavenged engine. This is really a very attractive feature.

### **Blower Scavenging with no Provision for Supercharge**

In this case the pressure drop has been estimated in a specific example for engine speeds of 1,000, 1,500, 2,000 and 2,500 r.p.m. Results indicate that the pressure drop at 1,000 r.p.m. is comparatively high. Here also, the pressure drop increases with increase in speed. This, however, can be offset to a certain extent by providing for supercharging.

The B.M.E.P. has been calculated for a blower scavenged engine having 90 mm. bore, (120 plus 120) mm. stroke, and working at 1,000 r.p.m. in the following cases: (1) Opposed piston version and (2) U-type version. The calculation shows that the U-type engine gives about 6 per cent less output than the opposed piston engine.

It is natural to compare the U-type engine to the opposed piston engine since both of them work with uniflow scavenging and have no valves. The U-type engine has all the advantages of the opposed piston engine. Further, it is simpler and more compact in construction since it requires only a single crankshaft to drive the pistons in the two cylinders. It can have a compact combustion chamber which is ideally suited for good air movement and proper spray distribution. This should result in a greater air utilisation and so a greater power output. It does not require—unless the combustion chamber is peculiarly shaped—special type injectors as in the case of the opposed piston engine. An ordinary injector with a hole type nozzle should serve the purpose.

Coming to disadvantages of this engine, simplicity is achieved only at the cost of a higher pressure drop and so a higher scavenge pressure and consequent reduction in the output of the engine. This, however, can be remedied at least partially by supercharging the engine. The U-type engine cannot be so well balanced as the opposed piston engine. Further, there is the problem of cooling the walls between the cylinders. This may, however, be solved by providing sufficient cooling spaces and greater flow of cooling medium.

### **Design of an Experimental Unit**

On the basis of the above theoretical investigation an experimental unit having the specifications given below has been designed to run primarily as a crankcase scavenged engine. With a few modifications it is proposed to run this unit later on as a blower scavenged engine.

### Design Specification

Cylinder bore: 3.543 in. or 90 mm.

Stroke (4.725 plus 4.725) in. or (120 plus 120) mm.

Displacement: 93.16 cu. in. or 1,527 cc.

Crankcase scavenged

Rating: 8 h.p. at 600 r.p.m. (estimated)

B.M.E.P.: 56.6 lb./sq. in. at 600 r.p.m.

### Materials

The materials chosen for this engine are mainly cast iron and carbon steel and are mostly available in India.

### Piston

It is a trunk type piston with a flat crown. Gudgeon pin bosses are reinforced by providing ribs. Material: Heavy duty cast iron. Three plain compression rings and one scraper ring are provided above the gudgeon pin bore and an extra scraper ring at the bottom of the skirt. Material: Pearlitic cast iron.

A fully floating gudgeon pin is retained in the piston by circlips. Material: 0.1 to 0.15 carbon case-hardening steel.

### Connecting Rod

The connecting rod shank is made round in section. The big end of the connecting rod is of the automobile type, split across the centre line of the crankshaft and held together by two high tensile carbon steel bolts. Material: 40 ton carbon steel.

The gudgeon pin floats in a phosphor bronze bush fixed in the small end of the connecting rod.

The big end-bearings are split bronze shells lined with white metal.

### Crankshaft

The double throw crankshaft has been made in two halves each having a crank-pin and a journal. The two halves are coupled together with a sleeve which has internal splines cut on it which mesh with splines on the respective halves. By this arrangement it is possible to vary the phase difference between the crank-pins from 5 to 20° in steps of 0.75°. Material: 40 ton carbon steel or heavy duty cast iron.

The two main bearings are bronze shells lined with white metal.

Bronze seal rings are provided at either end of the crankshaft to make the crankcase leak-proof and at the same time prevent lubricating oil supplied to the main bearings from coming into the crankcase.

Cast iron fly-wheels are provided at either end of the crankshaft.

### Engine Base

The base forms the bottom half of the crankcase. The shape of the crankcase closely follows the contour of the crankshaft so that the volume of the crankcase is a minimum. Separate main bearings caps have been provided. Material: Cast iron.

### Cylinder Block

Since the engine has been designed primarily as a crankcase scavenged engine, the upper half of the crankcase has been incorporated in the cylinder block casting, and inlet and exhaust ports are cast and no cylinder liners have been provided. The inlet ports are slightly inclined to the radial to give a swirl motion to the incoming air. But the exhaust ports are mostly radial.

Air enters the crankcase pump through two automatic flap valves, each having three spring steel flaps mounted on it.

### Cylinder Head

This contains mainly the combustion chamber which has been shaped specially to offer minimum resistance to flow of scavenge air. Material: Cast iron.

The fuel injector, placed at the centre of the combustion chamber, has a long stem nozzle with two holes. The angle between the holes is about  $110^\circ$ .

A decompression valve, which also acts as a safety pressure release valve, has been provided.

### Fuel Pump Drive

The fuel pump is operated by a double eccentric on the crankshaft to enable the study of the effect of injection timing and rate of injection on the performance of the engine.

The fuel pump and the injector are the only two parts of foreign make used in this engine.

An Acme forced-feed lubricator made in India, used for lubrication purposes, is operated by the same eccentric which drives the fuel pump.

### Cooling

The engine is water cooled.

### Lubrication

The main bearings are ring-lubricated. The crankpins are lubricated by Banjo rings into which lubricating oil is fed intermittently, under slight pressure, by the forced-feed lubricator. The cylinder wall's and gudgeon pins are to rely on splash lubrication. The fuel pump drive is lubricated by a drip-feed lubricator.

### Conclusions

Definite conclusions can be drawn only after an experimental investigation of the problem.

It is hoped to complete the fabrication of all the components of this engine in a very short period. After this, the engine will be assembled and tested for its performance characteristics. Further development work will be carried out in the light of these results.

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# Design Features of a Small Power Aero-Otto Engine

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The paper describes design details and the special features of an aero-engine single cylinder test unit of 25-30 B.h.p.

## 1. Introduction

A power plant of English manufacture has been proposed as propulsion unit for the Trainer Aircraft HT-2 of the Hindustan Aircraft Ltd., Bangalore. The first prototype HT-2 used a Gipsy Major 10 series engine of Messrs. De-Havilland and the second prototype as well as the series production HT-2 will use Cirrus Major 3 series engines of Messrs. Blackburn Engine Co. A scheme has been submitted to the C.S.I.R. jointly by Dr. Havemann, Professor and Head of the Internal Combustion Engineering Department, Indian Institute of Science and Dr. V. M. Ghatage, Chief Designer, Hindustan Aircraft Ltd., to develop an indigenous design of an aero-engine to replace the power plant of foreign origin as the propulsion unit for the HT-2. A six cylinder, in-line, inverted, air-cooled, direct drive, unsupercharged Otto-Engine of about 160 B.h.p. at 2,500 r.p.m. has been proposed and as an initial step the design of a single cylinder prototype of 25-30 B.h.p. at 2,500 r.p.m. has been taken up and is nearing completion. This work is being carried out with the help received from the Defence Science Organisation and its head, Dr. D. S. Kothari.

## 2. Design Specification

The choice of the type of engine has been dictated by a proposal to develop a replacement engine and, as such, the design of a single cylinder, inverted, air-cooled, direct drive, unsupercharged Otto-Engine of 25-30 B.h.p. running at 2,500 r.p.m. has been taken up.

The main features of the engine are as follows:

Cylinder Bore	...	...	...	4.130 in.
Stroke	...	...	...	4.520 in.
Displacement	...	...	...	60.7 cu. in.
Compression Ratio	...	...	...	6.0:1
Rating (max.)	...	...	...	25 to 30 B.h.p. at 2,500 r.p.m.
B.M.E.P. (max.)	...	...	...	138 lb./sq. in.
Piston speed (max.)	...	...	...	3,060 ft./min.

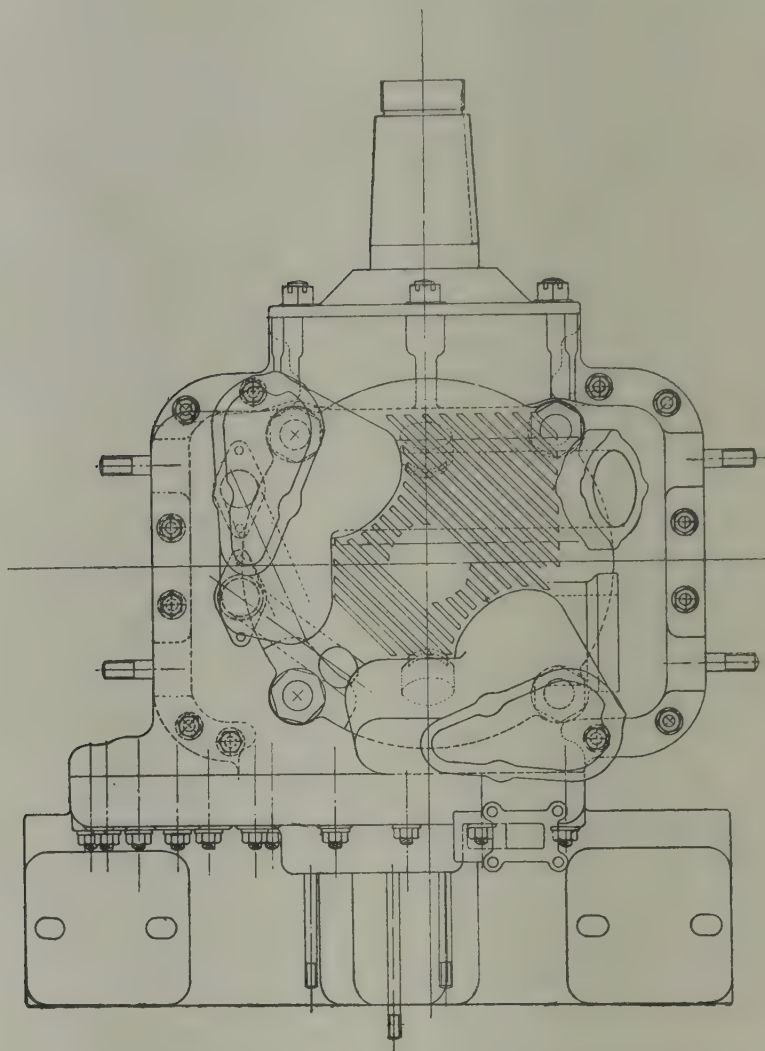


FIG. 3—PLAN OF THE SINGLE CYLINDER AERO-ENGINE

3.2. THE CYLINDER HEAD is proposed to be a sand casting in aluminum alloy. The Combustion Chamber is to have the shape of a segment of a sphere, so as to provide the most ideal shape.

3.21. VALVES, one inlet and one exhaust, are inclined at  $40^\circ$  to the vertical and are located in a plane  $45^\circ$  to the longitudinal axis of the engine. The horizontal fins are concentric with the cylinder bore and the vertical fins run parallel to the plane of the valves. This layout cuts down the cooling drag to a half of the conventional type designs, since the cooling air is turned through only  $90^\circ$  instead of  $180^\circ$ . The exhaust rocker housing, located parallel to the longitudinal axis, faces directly the entering cooling air, thereby providing the best possible cooling for the exhaust valve. The inlet rocker housing, located at right angles to the longitudinal axis, is at the trailing end of the cooling air, in effect providing conditions for a slightly warmer inlet, thereby cutting down the difficulties of cold start, fuel condensation, etc.

3.22. THE ROCKER ARMS are located in planes  $30^\circ$  to the horizontal and this layout satisfies the condition that no side thrust is experienced by the valves whilst in action,

3.23. THE VALVE GUIDES are of phosphor bronze and the Valve Seats of nickel-chrome high expansion steel are shrunk and peened into the roof of the head. The valve ports terminate in faced flanges on the port-side of the engine, that for the inlet being vertical whilst the exhaust flange is at an angle of  $30^\circ$  to the vertical. This arrangement allows both the inlet and exhaust manifolds to be mounted on the same side of the engine.

3.24. Dual ignition is provided and the plugs are located symmetrically around the inlet and exhaust valve ports, inclined at  $30^\circ$  to the horizontal.

3.25. The apex of the spherical combustion chamber is easily accessible and a bored facing gives the fixture for either an injector or a pressure pickup.

3.26. The interspaces between the horizontal fins are made solid, providing solid lugs for tightening down the cylinder head to the barrel and crankcase. The horizontal fins are trimmed to accommodate the exhaust push rod tube. The inlet push rod is passing through the material of the head and hence the inlet push rod tube is made in two sections, one above and one below the horizontal fins.

3.3. VALVES AND VALVE OPERATING GEAR—The valves are of the conventional type and the inlet valve head is slightly larger in diameter. The valve seats and tips are stellited. The valves are operated by conventional arrangement of tappets, push rods and rocker arms worked by a camshaft running in bearings on the star board side of the front and rear walls of the crankcase. The camshaft gear is attached to the camshaft through an arrangement of keyways designed to permit accurate valve timing.

3.31. The lift of the cams is transmitted to the valves through tappets working in aluminium alloy guides, spigotted into the crankcase and secured by studs and nuts; solid push rods of steel with steel ball ends and rockers mounted on ball bearings over spindles fixed to the rocker housings. The push rod tubes are of light alloy. The rocker mechanism is enclosed by light alloy cast covers, which act as a sump which permits the working parts to dip into the oil bath.

3.4. The Piston will be machined from an aluminium alloy sand casting. The gudgeon pin bosses are attached to the crown of the piston by solid webs so that the thrust is transmitted directly to the pin. The fully floating gudgeon pin is retained in the piston by two retainer spring clips expanding into recesses in the piston, one at either end. Three pressure rings are fitted in grooves machined between the crown and the gudgeon pin. A scraper ring is fitted in the groove machined in the skirt and the oil is drained through holes drilled at an angle through the bottom of the ring groove.

3.5. The CONNECTING ROD is of a uniform I-section, machined all over from a nickel-chrome steel forging, the big end being split across the crank pin centre-line and clamped together by two bolts and nuts. The big end bearing is steel backed, white metal lined, the connecting rod half being located by a dowel stud in the rod. The small end of the rod is fitted with phosphor bronze bush.

3.6. The single-throw CRANKSHAFT is machined from a nickel-chrome steel forging and is carried in two main bearings which are of steel backed,

white metal lined type. The crankpin is hollow, the bores being connected by drillings through crank webs to provide a passage for high pressure oil to lubricate the big end bearing. The ends of the bore are blanked off by sealing cups clamped to the crank webs by transverse bolt and nut. The forward journal is hollow, being interconnected to the crankpin by drilling for high pressure oil and the end is blanked off by drawing a sealing cup 'taught' by a stud bolt. The hollow rear main bearing journal is keyed internally to receive the hub of the crankshaft gear wheel. A revolution counter attachment can be provided at the rear end of the crankshaft gear wheel. At the beginning of the front main bearing journal, a ball thrust bearing is fitted to locate the shaft and to transmit the fore and aft thrust loads. At the front the crankshaft runs into a slow taper to receive the fly-wheel and a key, set in parallel to the taper. Drive is taken off through six bolts attached to the fly wheel hub boss. The webs of the crankshaft are extended and balanced weights are attached to the extended ends.

3.7. THE CRANKCASE AND TOP COVER are sand castings in aluminium alloy bolted together in the horizontal plane of the crankshaft centre line. The crankshaft main bearing lower halves are machined out of bosses supported by the front and rear walls of the crankcase. The extension of the front main bearing boss is bored to form the lower half housing of the thrust ball bearing. The main bearing shells are retained by separate caps by two studs and nuts.

3.71. The underside of the crankcase is faced and bored to take the cylinder barrel spigot and tappet guides for their respective hold down studs. On the star board side a facing suitably bored, serves as a mounting for the fuel pump secured by two studs and nuts.

3.72. The rear wall of the crankcase projects below the level of the cylinder facing and a depression in the casting forms a rim, shaped to conform with the profile of the rear cover which is fitted to the machined face of the rim and secured by a series of studs and nuts. A boss formed on the rear wall is bored to receive the idler gear spindle and a drilling between the boss and the main bearing provides an oil passage for the lubrication of the idler gear bearings. A horizontal oilway drilled through the crankcase rear wall supplies high pressure oil to the bearings. Rectangular facings on the sides provide the attachment points for mounting.

3.73. THE TOP COVER is secured by a series of bolts and nuts. A boss at the forward end is machined to form the top half of the thrust ball bearing housing, whilst the rear end is faced vertically to take the rear cover.

3.8. REAR COVER, TIMING GEARS AND ACCESSORIES DRIVE—The rear cover is secured to facing on the rear of the crankcase and top cover. External mountings are provided for magnetoes, oil filter and oil pump unit. The gear mounted at the rear of the crankshaft provides the drive for the camshaft through an idler gear mounted on the idler spindle attached to the crankcase. The camshaft drive gear meshes with another spur gear which drives the high pressure gear type oil pump. The idler gear boss is extended to form a sleeve and is flanged to which is attached a helical gear mounted on a short transverse shaft, running on ball bearings, and having flexible vernier couplings on

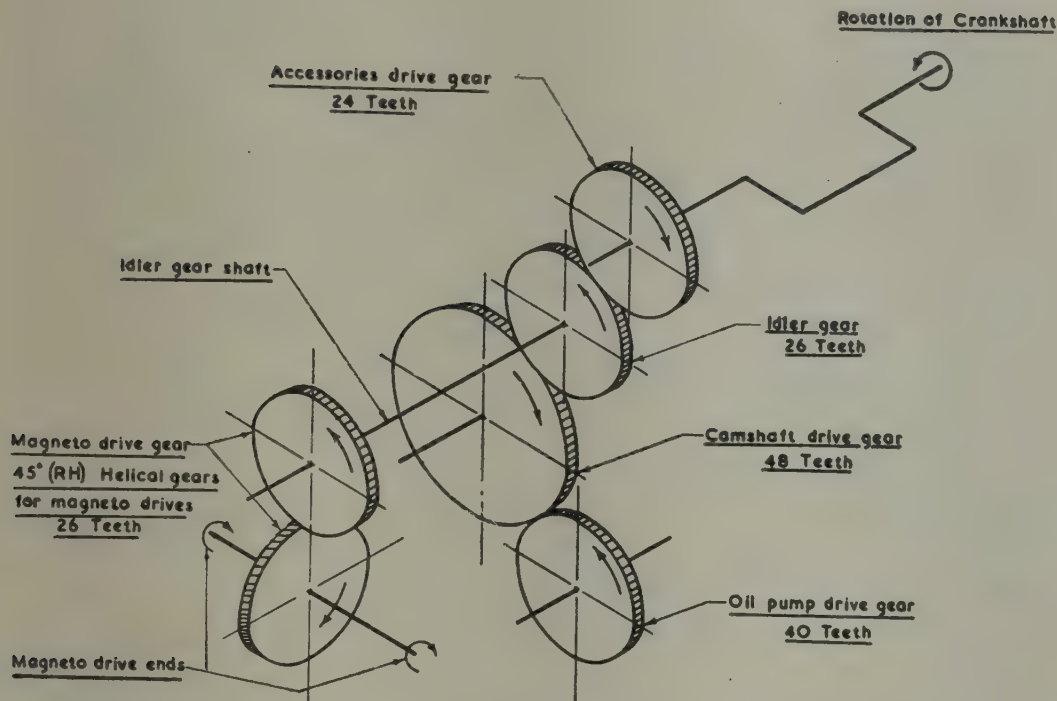


FIG. 4—SCHEMATIC GEAR TRAIN DIAGRAM OF ACCESSORIES DRIVE

either end, to form the drive for the magnetoes mounted on platforms integral with the cover. An impulse starter is incorporated with the star board magneto coupling. The control levers are mounted on the rear cover interconnecting throttle and ignition control.

3.9. LUBRICATION of the engine is on the dry sump principle with the high pressure oil pump and pressure and suction filters, mounted in the rear cover. The high pressure oil pump is of the gear type. Oil drains to the tank and is drawn by the pump through a gauze suction filter and delivered through the Autoklean pressure filter-mounted horizontally in the centre of the rear cover, to the oil pipe connecting the oil way in the crankcase. From there oil is led through drillings to lubricate the main bearings and big end bearing from which oil is projected from holes drilled in the connecting rod caps on the cylinder walls and camshaft. An oil pipe from the pressure filter housing connects to a union in the rear cover which provides an oil jet spray to the helical gears and the splash lubricates the camshaft gear and the oil pump gear. Oil drainage collecting in the wells formed by the projection of the cylinders inside the crankcase is drained to the rear cover, from where oil drains to the oil tank.

The valve rocker gear is lubricated separately by splash from rocker covers which are fitted with oil to a prescribed level.

3.91. CARBURATION will be adopted on the initial run of the engine. Fuel injection direct into the cylinder will be finally adopted to give the increased output ranging from 6 per cent to 15 per cent.

3.92. COOLING is effected by air which enters the open end on the star board side of the front scoop. As already set out elsewhere, the cooling air is

diverted through  $90^\circ$  only and it is anticipated that the cooling drag will be cut down to a half of the drag of similar units.

4. MATERIALS—The design takes into account fully the materials which are available in this country. S.A.E. specifications have been adopted so far, as a guide, and during the fabrication equivalent materials, which could either be procured in this country or be fabricated from known analysis, would be used.

### Conclusion

The problem of developing an engine design is an extremely complex one. A few characteristic features have been incorporated in the design to give the unit an individuality, though the basic design is based on similar established units.



# Development of Facilities for Testing Gas Turbines and Gas Turbine Components in the I.C.E. Department

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## 1. Introduction

The Ministry of Supply, London, very kindly made available in February 1950 to the Department of Internal Combustion Engineering, Indian Institute of Science, the turbo-jet Derwent-V engine manufactured by Messrs. Rolls Royce Ltd., Derby. About the same time, the Ministry of Education of the Government of India also gave financial aid to start instruction and create facilities for research in Gas Turbines.

The Derwent-V turbo-jet engine has been installed for test and instructional purposes. The testing facilities built up for this unit enables the Department to conduct tests on other similar engines also. The Department has also undertaken to design and to develop an indigenous gas turbine of about 250 s.h.p. Various test set-ups for development of the components have therefore been planned and are being constructed.

## 2. Testing of Piston Engine and Gas Turbine Compared

The mode of testing a gas turbine is quite different from that of conventional piston engine. In the latter the main thermodynamic operations such as compression, heat addition, expansion occur in a single phase—the cylinder, and any alteration which might be an improvement in one direction is bound to have a close effect on other parts of the cycle. The engine is therefore to be tested as a whole. In the gas turbine, on the other hand, the several operations of the cycle are distributed among a number of components like the compressor, the combustion chamber and the turbine, which are in themselves complete units capable of performing individual operations. Each component can therefore be tested individually and the effect of changes carried out on any component on the performance of the engine as a whole can be gauged.

Secondly, while the performance of the reciprocating engine is largely affected by adjustments on carburettor, ignition timing, jacket temperature, etc., made on the test bed, relatively little can be done on the test bed to affect the performance of a gas turbine.

### 3.1. Performance of the Derwent-V

It will be noted that the maximum thrust of 3,500 lb. is given at 14,700

r.p.m. with a jet temperature of 675°C. and compressor delivery pressure of 50 lb. at 15,000 r.p.m.

### 3.2. Normal Test Measurements

The various measurements which are usually taken during a test for a complete analysis of the engine are:

- (a) Measurements of overall performance
  - (i) speed, (ii) thrust, (iii) fuel consumption
- (b) Air mass flow measurements, 63.2 lb./sec.
- (c) Temperature measurements
  - (i) ambient atmospheric temperature, (ii) compressor delivery temperature, (iii) exhaust jet temperature, (iv) miscellaneous temperature associated with mechanical features
- (d) Pressure measurements
  - (i) ambient atmospheric temperature, (ii) compressor delivery pressure, (iii) turbine inlet pressure, (iv) exhaust jet pressure, both pitot and static

## 4. Engine Mounting

The Derwent-V engine is a very light machine (1,280 lb.) for its output and is a simple rotary engine free from heavy vibrations. Consequently, the foundations required for ground mounting and running need not be very massive. For thrust measurement, the engine mounting must be such as to allow the engine to move freely in the direction of thrust. The engine therefore can be mounted on a cradle suspended from the roof. The movement of the engine need not be more than 0.06 in. and is prevented by a thrust measuring instrument which indicates the thrust developed.

A test stand similar to the Rolls-Royce type has been fabricated here. The engine is mounted on a cradle stand on two spherical bearings situated at approximately the horizontal diameter of the compressor casing. The cradle is suspended by means of four flat spring strips attached to the inside face of four built up columns. The fore and aft columns are connected by diagonal bracing. The right hand and left hand columns are connected together by an adjustable spacer block with bolts and nuts. The stand so formed is fixed on to two cross rails made of cast iron with T slots. The cross rail bed plates are mounted on two other similar bed plates situated on either side parallel to the longitudinal axis of the engine. The longitudinal bed plates are fixed in the ground by foundation bolts.

A similar bed plate arrangement in front of the engine supports a rigid structure on which the thrust measuring unit is mounted, at the appropriate height to prevent the engine cradle from moving forward by means of a pressurised thrust meter which holds an axial rod against the engine cradle.

The cradle is provided with rear brackets to support the rear engine suspensions,

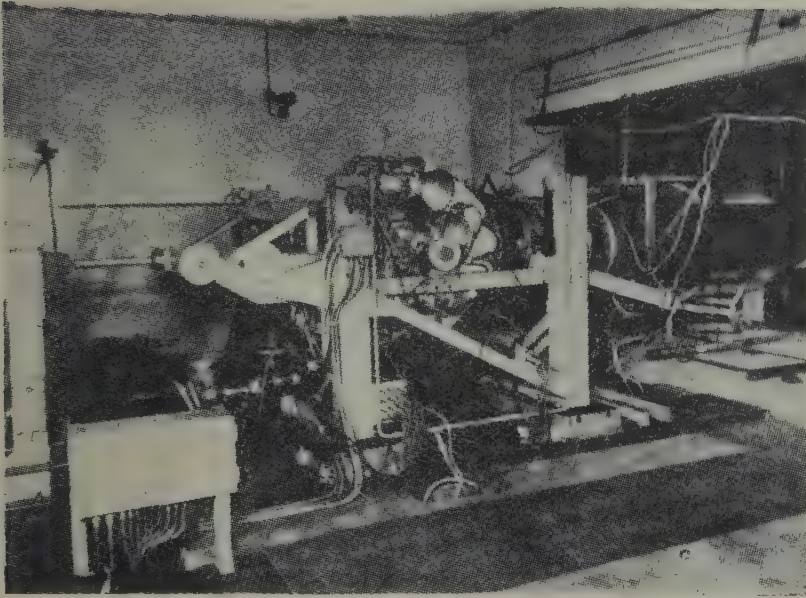


FIG. 1—TEST STAND FITTED WITH GAS TURBINE

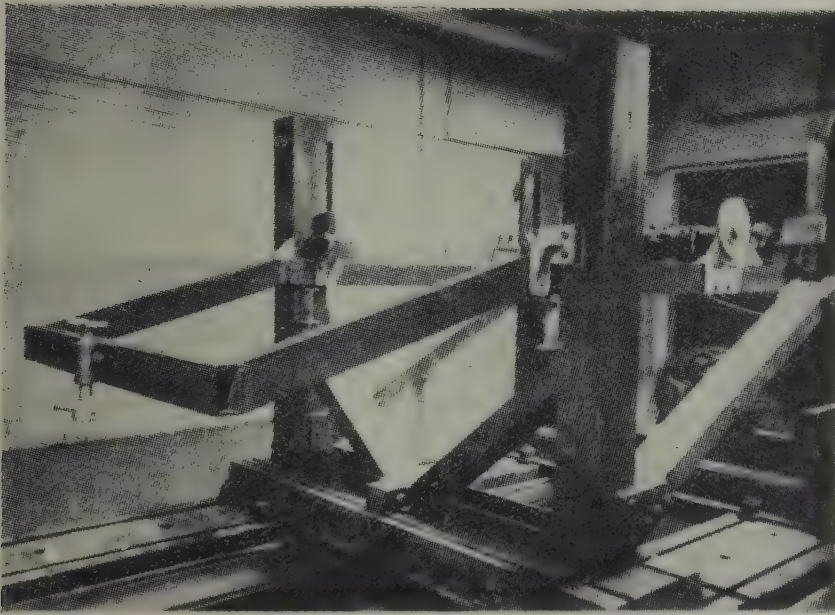


FIG. 2—TEST STAND

## 5. Test House

In view of the high speed of the engine and the tremendous energy stored up in these engines which might be released at any moment if a major failure should occur, the engine room or 'test cell' should always be isolated from the observation and control room. On account of the excessive noise of the engine the observation room and the test cell have to be isolated for noise. Large quantities of exhaust gases evolved by the engine must be diffused into the atmosphere.

A room of 48 ft.  $\times$  10 ft. 6 in. has been partitioned by a double wall incorporating an air gap so that an observation and control room of 12 ft.  $\times$  10 ft. 6 in. is created. The available height of the room is 16 ft. which has been divided up with a ceiling at 12 ft. and the duct serves as a passage for the air and allows air measurements with the help of a venturi placed in the passage. Suitable air deflectors have been placed to divert the air to the engine which is mounted below opposite the observation room. The exhaust of the engine is blown into a 50 ft. long 30 ft. diameter concrete pipe line placed horizontally and coaxial with the engine at a height of 3 ft. 6 in. The gases

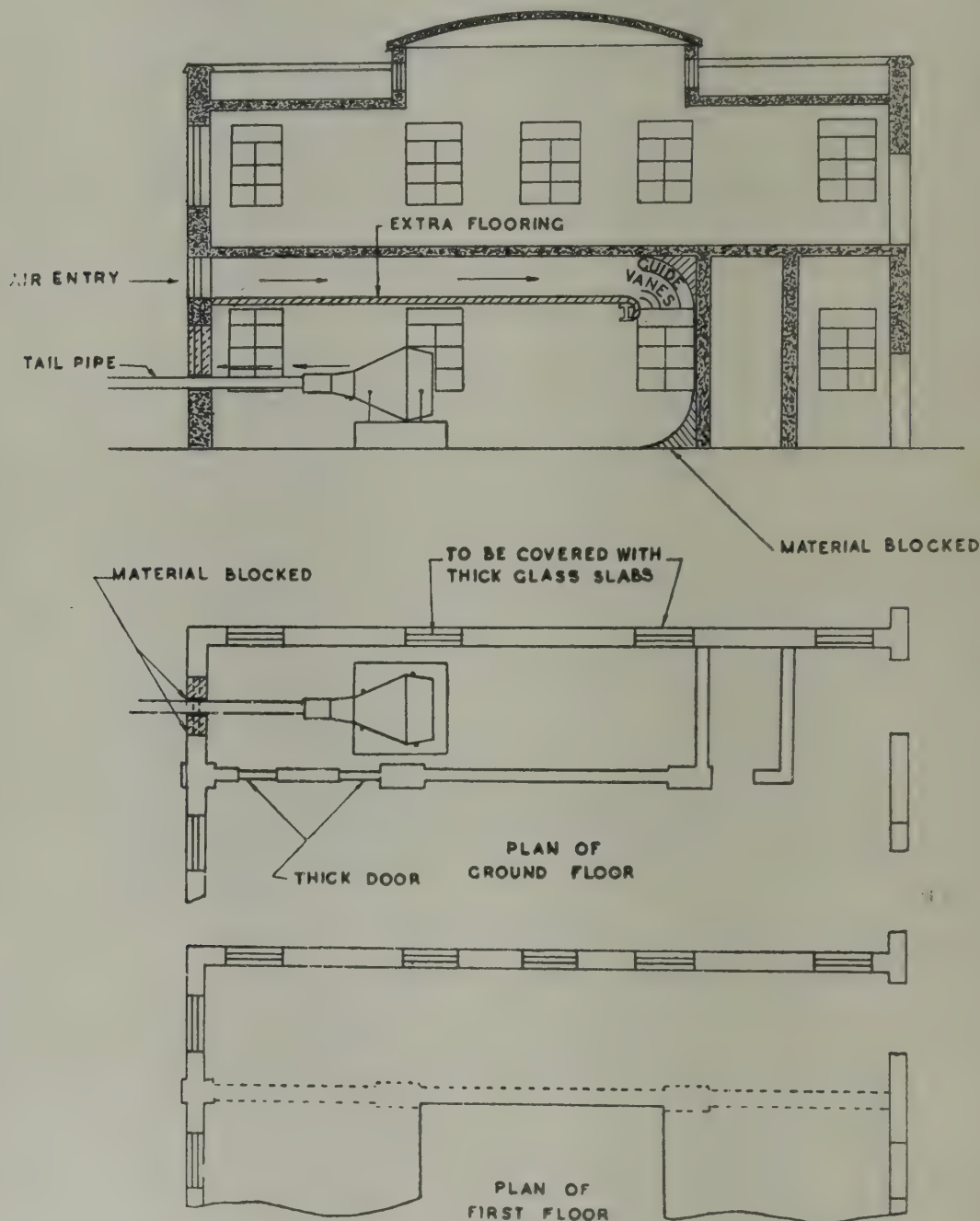


FIG. 3—DETAILS OF TEST HOUSE

escaping through the pipe line are directed upwards by a masonry deflector placed at the end of the pipe line.

All openings in the room such as doors and windows are provided with steel shutters to be closed when the engine is working and air measurements are to be taken.

The control room is equipped with instruments for taking readings as listed under 3.2. and estimating the performance of the engine. The starting panel and batteries required for starting the engine are situated in the engine room.

## 6. The Fuel System

The fuel is pumped from a 600 g. capacity tank in an underground masonry pit by a centrifugal pump to an overhead tank with an overflow return. Fuel from the overhead tank is taken through filters, sight gauge, flow meter all situated and controlled from the observation room.

The engine room is provided with a tank of about 10 gal. capacity for inhibiting the engine for storage.

An oil tank for lubricating oil is also provided to supply the engine as a slave unit, when the engine tank is isolated for test runs. The oil may be cooled by a cooler provided for the purpose.

The installed fuel tank serves at present for test not beyond one hr's. run. A larger capacity tank would be installed at a later date.

## 7. Component Testing

Component development requires testing facilities for studying the performance of each component individually. Each component test bed requires a different instrument technique.

The gas turbine plant that is being developed in this department is about 250 s.h.p. A gas turbine of this capacity has to be run at about 20,000 r.p.m. The compressor required in such a unit absorbs about 500 h.p. The driving turbine should therefore develop about 750 h.p. The combustion chambers have to handle a mass flow of 5 lb./sec.

### 7.1. Combustion Chamber Test Rig

A test rig has been set up for testing a combustion chamber for half a pound of air per second. The apparatus consists of a Vane type 4 in. Hamworthy rotary compressor delivering 360 cu.ft. of air per minute at 15 lb. gauge to supply air to the combustion chamber through a venturi. A by-pass valve is used to control and measure the air mass flow. An engine driven centrifugal supercharger is also being put in parallel with the above compressor. Fuel is supplied from a reservoir by a six cylinder, C.A.V. diesel pump directly driven by a  $\frac{1}{2}$  h.p. variable speed motor. A flow meter is incorporated on the low pressure side to determine the fuel consumption. A pressure gauge near the burner indicates the delivery pressure.

The ignition of the air-fuel charge is by an acetylene torch igniter with a spark plug energised by a booster and a battery.

The above set up can measure (a) the flame length, (b) the flame temperature, (c) the velocity and pressure and temperature distribution at inlet and outlet, (d) the combustion efficiency, (e) the pressure loss and (f) the limits of mixture blow outs.

### 7.2. The Compressor Test Bed

The compressor test bed consists of a marine packard 12 cylinder V petrol engine of 1,200 h.p. running at 2,400 r.p.m. The drive is taken through a Voith's step up gear of 1.10 gear ratio.

### 7.3. The Turbine Test Bed

The turbine test bed consists of a 1,000 h.p. water brake on order from Germany to which the turbine may be coupled directly. At the first instance it is proposed to run the turbine with the exhaust gases of the marine engine by throttling the exhaust gases to the required degree of pressure.

### 7.4. Cascade Wind Tunnel

A small wind tunnel is being designed by testing the performance of turbine blade cascade. For this purpose the super-charger in the Rolls Royce Griffin Engine is being utilised as a suction blower. A Schleren apparatus is being obtained for the study of gas flow round the cascade of blades.

## Conclusion

With the completion of the test rigs mentioned, it is hoped the Department will be fully equipped to impart instruction, conduct researches and to develop successfully the contemplated indigenous gas turbine.

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# The Function of Expansion Ratio in Determining the Cyclic Efficiency of I.C. Engines

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Consider the two basic cycles, i.e., the constant volume and constant pressure cycle. In the constant volume cycle (Fig. 1), the inlet charge is compressed adiabatically from point 1 to 2. At 2 heat  $H_2$  is added at constant volume to point 3, then adiabatic expansion takes place from point 3 to 4 and heat  $H_3$  is rejected from 4 to 1. Therefore the thermal efficiency of the cycle is

$$E = \frac{H_2 - H_3}{H_2} = 1 - \left(\frac{1}{r}\right)^{\gamma-1}$$

where  $r$  = expansion ratio and  $\gamma$  is ratio of specific heats of perfect gas.

Similarly, the thermal efficiency for the constant pressure cycle is

$$E = \frac{H_1 - H_2}{H_1} = 1 - \left(\frac{1}{r}\right)^{\gamma-1}$$

In both these cases the expansion ratio is equal to the compression ratio. Therefore in these cycles any of the two ratios can be used.

In diesel cycle (Fig. 2) the air is compressed adiabatically from point 1 to 2 and heat  $H_1$  is added between 2 and 3 at constant pressure, adiabatic expansion takes place from 3 to 4 and heat  $H_2$  is rejected at constant volume from 4 to 1.

$$\text{The efficiency is } E = \frac{H_1 - H_2}{H_1} = 1 - \left(\frac{1}{r}\right)^{\gamma-1} \left[ \frac{r_c^\gamma - 1}{r(r_c - 1)} \right]$$

where  $r_c$  is the cut off ratio or the ratio of volume at the end of compression and volume at the end of heat reception.

$$\begin{aligned} r_c &= \frac{V_3}{V_2} \cdot \text{But } V_3^\gamma = \frac{P_4}{P_3} \cdot V_4^\gamma \\ \therefore V_3 &= \left(\frac{P_4}{P_3}\right)^{1/\gamma} \cdot V_4 \text{ or } \frac{P_4}{P_3} = \left(\frac{V_3}{V_4}\right)^\gamma \\ \therefore r_c &= \frac{V_3}{V_4} \cdot \frac{V_4}{V_2} \cdot \text{But } V_4 = V_1 \\ &= \frac{V_3}{V_1} \cdot \frac{V_1}{V_2} = \frac{r}{r_{ex}} \text{ where } r_{ex} = \text{expansion ratio.} \end{aligned}$$

$$\therefore E = 1 - \left(\frac{1}{r}\right)^{\gamma-1} \times \left[ \frac{\left(\frac{r}{r_{ex}}\right)^{\gamma} - 1}{r \left(\frac{r}{r_{ex}} - 1\right)} \right]$$

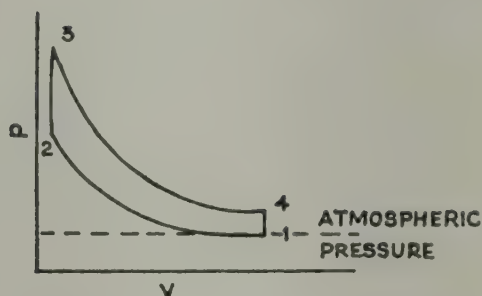


Fig. 1

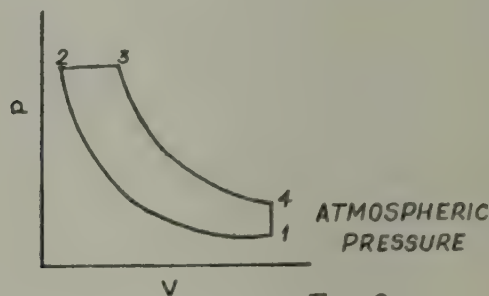


Fig. 2

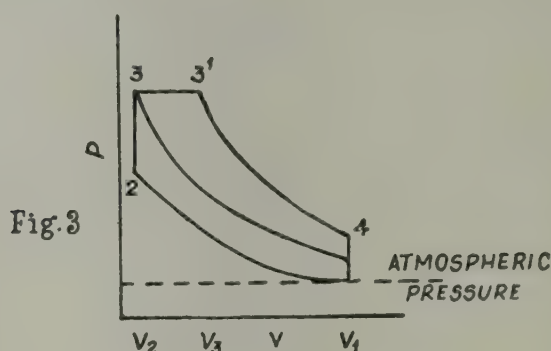


Fig. 3

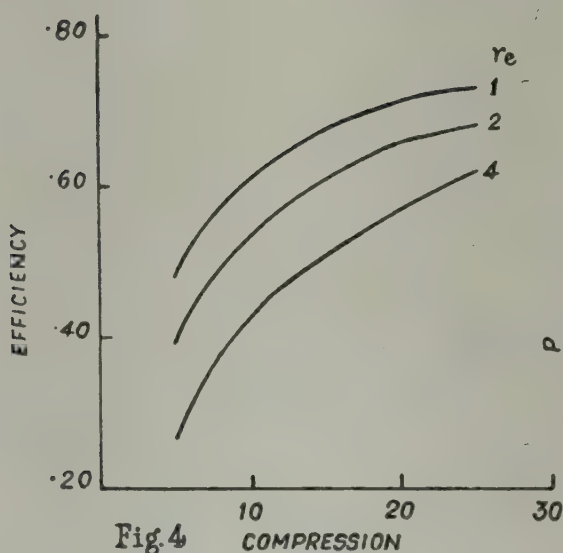


Fig. 4

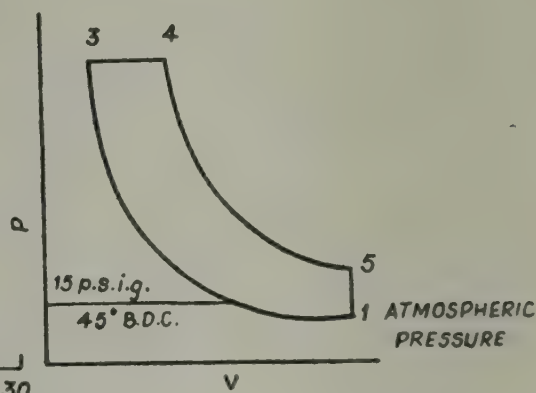


Fig. 5

FIG. 1—CONSTANT VOLUME CYCLE

FIG. 2—DIESEL CYCLE

FIG. 3—MIXED CYCLE

FIG. 4—COMPRESSION RATIO *vs*  
EFFICIENCY AT  $R_0$  EQUAL TO 1, 2  
AND 4

FIG. 5—MILLER CYCLE

The above expression for the thermal efficiency of the diesel cycle is the same as the one for the constant pressure cycle except for the minus factor

multiplied by  $\frac{\gamma}{r(r_c - 1)}$  which is always greater than unity. In the second expression for the thermal efficiency of the cycle it has been shown that this factor is governed by the expansion ratio  $r_c$  which is never greater than the compression ratio, and is never less than unity.

Since  $\frac{\gamma}{r(r_c - 1)}$  increases as  $r_c$  increases, we have the value of thermal efficiency decreasing as  $r_c$  increases.

All the cycles discussed above are not practical. Time element is the major factor and therefore these cycles can only be approximated in practice. The mixed cycle in Fig. 3 represents a practical cycle where part of the heat is added at constant volume and part is added at constant pressure. As shown in Fig. 3 this cycle is split into the constant volume and the diesel cycles. We have already derived the expression for the above cycles, therefore we get the thermal efficiency of the mixed cycle

$$E = \frac{H_v E_v + H_b E_b}{H_v + H_b} \quad \text{where } H_v \text{ is the heat added at constant volume and } H_b \text{ is the heat added for diesel cycle}$$

$$\text{or } E = 1 - \left(\frac{1}{r}\right)^{\gamma-1} \left[ \frac{p \cdot r_c - 1}{(p-1) + p \cdot r(r_c - 1)} \right] \quad \text{where } p = \frac{P_3}{P_2}$$

The above equation shows that the expansion ratio only affects the diesel cycle part of the mixed cycle, therefore our purpose will be served by considering the diesel cycle alone. The efficiency is

$$E = 1 - \left(\frac{1}{r}\right)^{\gamma-1} \times \frac{r_c - 1}{r(r_c - 1)}$$

In Fig. 4 the efficiency of the diesel cycle is plotted against the compression ratio for  $r_c$  equal to 1, 2 and 4. From the figure, it is apparent that the efficiency of the diesel cycle increases as  $r_c$  approaches unity. Since the above factor is governed by the expansion ratio the efficiency will increase as the expansion ratio increases. Therefore, if a cycle can be developed with an expansion ratio greater than the compression ratio, we will get higher efficiency.

Several attempts have been made to develop a cycle which has an expansion ratio higher than the compression ratio, but so far no one has been successful.

An eminent diesel engine designer, Ralph E. Miller, has developed a cycle which approximates to this condition. Miller cycle is based on introducing the inlet air at 15 p.s.i.g. after intercooling it to 100°F. and shutting off the inlet valve 45° before B.D.C. so that at B.D.C. the pressure in the cylinder is 6 p.s.i.g. and temperature is approx. 50°F. Miller gives various advantages of this cycle but we will here consider only the increase in the

expansion ratio of the cycle. From Fig. 5 it will be clear that work involved in expanding to 6 p.s.i.g. and compressing the inlet charge back to 16 p.s.i.g. is negligible. Therefore, for all practical purposes the compression starts  $45^\circ$  after the B.D.C. and the expansion is carried  $45^\circ$  more than the compression.

The following Table gives a comparison between an ordinary supercharge diesel engine and a diesel engine run on Miller cycle. The gain in thermal efficiency is clearly shown by the two specific fuel consumptions, i.e., 0.375 lb. B.h.p. hr. for ordinary cycle as compared to 0.365 lb. B.h.p. hr. for the other cycle. There are many factors which contribute to this increase in efficiency but for the present we will only consider the effect of expansion ratio.

TABLE—COMPARISON BETWEEN SUPERCHARGE DIESEL ENGINE AND A MILLER CYCLE DIESEL ENGINE

			Conventional Turbocharged (non-intercooled)	Superthermal Engine
Engine Bore and Stroke (in.)	...	...	$13 \times 16\frac{1}{2}$	$13 \times 16\frac{1}{2}$
Revolutions per min.	...	...	450	450
Piston speed	...	...	1237.5	1237.5
Maximum firing pressure (p.s.i.g.)	...	...	850	850
B.h.p. for continuous service (per cyl.)	...	...	150	200
Brake mean effective pressure (p.s.i.)	...	...	120	160
Ambient temp. ( $^\circ\text{F.}$ )	...	...	90	90
Air temp. at supercharge discharger ( $^\circ\text{F.}$ )	...	...	142	250
Air temp. to intercooler ( $^\circ\text{F.}$ )	...	...	—	250
Water temp. to intercooler ( $^\circ\text{F.}$ )	...	...	—	90
Air temp. after intercooler ( $^\circ\text{F.}$ )	...	...	—	100
Air temp. entering engine ( $^\circ\text{F.}$ )	...	...	142	100
Air temp. at beginning of compression ( $^\circ\text{F.}$ )	...	...	142	50
Air pressure at supercharger discharge (p.s.i.g.)	...	...	4	15
Air pressure entering engine (p.s.i.g.)	...	...	4	15
Air pressure at beginning of compression (p.s.i.g.)	...	...	4	6
Heat to cooling water				
B.t.u./min./B.h.p.	...	...	16.20	12.15
Total B.t.u./min. at above rating (per cyl.)	...	...	2430	2430
Heat to lubricating oil...				
B.t.u./min./B.h.p.	...	...	6.66	5.80
Total B.t.u./min. at above rating (per cyl.)	...	...	1000	1160
Heat removed by intercooler				
B.t.u./min./B.h.p.	...	...	—	8.2
Total B.t.u./min. at above rating (per cyl.)	...	...	—	1640
Typical Comparable Fuel Consumption Rates on Dynamometer test lb./B.h.p./hr.				
Full Load	...	...	0.375	0.369
$\frac{3}{4}$ Load	...	...	0.37	0.355
$\frac{1}{2}$ Load	...	...	0.375	0.365
Mechanical Efficiency at full load (6 cylinder engine)			83.2%	85.4%

In conclusion, a study of this mixed cycle would help the engine designer considerably, if indicator diagrams are studied for various cycles at different speeds and loads. This data should be plotted as in Fig. 4 for a particular speed. From these charts a designer could determine the condition which will suit his purpose best.

# Production of Precision Forging

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Production of precision forging such as connecting rods, crankshafts, camshafts, gear blanks, valves, etc. is one of the important factors in the development of internal combustion engine industry. Steel or light alloys forging used in internal combustion engines built for industrial purposes, aeroplanes or motor vehicles are usually produced in the same works together with various forgings required by other branches of industry especially the motor vehicle industry.

To cover the rapidly growing demand for precision forgings and to produce them cheaply, forging works must adopt modern production methods. The forging shop is no more an obscure smoky place—it is a modern, well equipped shop combined with a large tool room containing special machine tools.

The greater part of the precision forgings consumed by motor and aircraft industries is produced on hammers usually known as drop hammers or on horizontal upsetting machines known as forging machines and only a small percentage is made on presses. The drop hammer is far more extensively used than any other machine. The forging methods used on the drop hammers may be divided into 2 groups:

1. Forging on 2 separate hammers, of which one is auxiliary ; the latter is used only for preparing the piece of metal prior to final forging. The other—the drop hammer—is for final forging known as die forging or die stamping.
2. Forging on one hammer only which is equipped with multi-impression special dies designed in such a way that a complete sequence of forging operations may be carried out on one hammer.

The first set-up is used for small production and the second method is preferred for large output when thousands of forgings are required. The forging on 2 separate hammers requires inexpensive set of tools. The auxiliary hammer is usually equipped with a simple open anvil (Fig. 1a) and the drop forging hammer is provided with a simple single impression die (Fig. 1b). Both the hammers are installed side by side so that complete forging can be carried out from the same heat.

The forging on one hammer only is made with the use of multi-impression dies (Fig. 1c) which apart from having the final impression, have usually several auxiliary ones. This modern method when applied in the production of small and medium forgings gives 2 to 3 times greater output than the forging on 2 separate hammers, but good results greatly depend on the design of the die. The problem is rather difficult. At the moment there is

very little information available in technical literature concerning this matter and the practical knowledge more or less is confined to up-to-date manufacturers of precision forgings. It is hoped that all the technical information contained in this paper may assist the die and tool designer in handling new production problems.

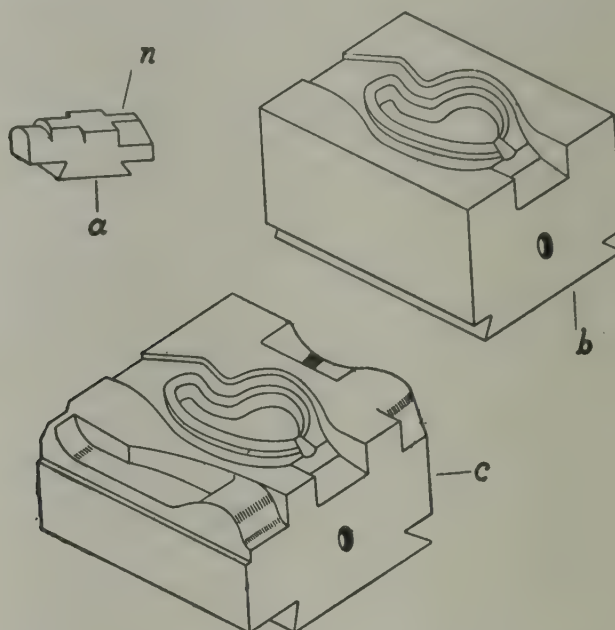


FIG. 1—FORGING AND DROP FORGING DIES USED FOR THE PRODUCTION OF DROP FORGINGS: (a) Open anvil for auxiliary hammer with cut-off impression (n) ; (b) Single impression drop forging die ; (c) Multi-impression drop forging die

## DIE AND TOOL STEELS—THEIR APPLICATION AND HEAT TREATMENT

### Die Blocks

Die blocks used for the production of forging dies are manufactured from high grade special steels made by the open hearth process either basic or acid or by electric furnace. The ingots are worked under the large forging presses, varying from 2,000-4,000 tons after which operation the die blocks are normalised, and usually heat treated in the final stage by quenching and tempered to the required hardness.

### Composition and Heat Treatment

Composition and heat treatment of die blocks depend on the type of the die and its application. Typical steels used for the production of forging dies and tools are given in Table 1.

### Application of Steels

Application of steels listed in Table 1 is briefly described in Table 2.

TABLE 1—TYPICAL STEELS USED FOR DIE BLOCKS, DIE INSERTS AND TRIMMING TOOLS

Steel	Type of steel	% C	% Mn	% Si	% Cr	% Ni	% Mo	%	Anneal- ing °C	Harden- ing °C	Quench- ing medium
DIE BLOCKS AND DIE INSERTS											
A	Carbon steel	0.55-0.65	0.50-0.80	0.80 max.	—	—	—	—	770-800	810-830	Water
B	Nickel-chromium steel	0.50-0.60	0.50-0.80	0.30 max.	0.50-0.80	1.50-1.70	—	—	760-790	810-830	Oil
C	Nickel-chromium-molybdenum steel	0.50-0.60	0.50-0.80	0.30 max.	0.50-0.80	1.25-1.70	0.25-0.30	—	760-790	820-840	Oil
D	Chromium-molybdenum-vanadium steel	0.50-0.60	0.70-0.90	0.30 max.	0.85-1.05	—	0.45-0.50	V = 0.06-0.012	760-790	830-850	Oil
E	Chromium-molybdenum steel	0.18-0.23	0.40-0.60	0.30 max.	1.40-1.60	—	0.45-0.55	—	770-800	850-870	Oil
F	Nickel-chromium-molybdenum-vanadium steel	0.50-0.60	0.50-0.80	0.30 max.	0.75-1.00	2.00-2.25	0.70-0.90	V = 0.08-0.012	760-790	820-840	Oil
G	Special steel for nitriding	0.45-0.55	0.40-0.65	0.30 max.	1.40-1.80	2.25 max.	0.10-0.25	Al = 0.90-1.30	790-820	880-900	Oil
H	Tungsten-chromium-vanadium steel	0.35-0.45	0.20-0.40	0.30 max.	3.00-3.50	—	—	W = 17.00-18.00 V = 0.10-0.15	860-890	slow-850 1080-1100	Air
I	Tungsten-chromium-nickel-molybdenum-vanadium steel	0.28-0.34	0.20-0.50	0.30 max.	3.00-4.00	3.00-4.00	0.30-0.60	W = 7.25-8.25 V = 0.40-0.60	860-890	slow-850 1140-1160	Oil
TRIMMING, PUNCHING, AND STRAIGHTENING TOOLS											
J	Group of carbon steel	0.50-1.50	0.20-0.40	0.25 max.	—	—	—	—	10-20 Above Ac <sub>3</sub>	30-50 Above Ac <sub>3</sub>	Water or Oil
K	Manganese steel	0.85-0.90	1.80-2.00	0.35 max.	—	—	—	—	750-780	770-800	Oil
L	Chromium-vanadium steel	0.75-0.85	0.20-0.40	0.30 max.	0.70-0.90	—	—	V = 0.20-0.30	750-780	770-800	Oil
M	Tungsten-chromium-vanadium steel	0.40-0.50	0.30-0.40	0.60-0.80	1.25-1.40	—	—	W = 2.20-2.40 V = 0.10-0.20	770-800	850-870	Oil

TABLE 2—APPLICATION OF STEELS

A	Carbon steel	Dies for drop hammers and forging machines when production is small (few hundreds of forgings only)
B	Nickel-chromium steel	Dies for drop hammers when production is not exceeding 2,000-3,000 forgings
C	Nickel-chromium-molybdenum steel	Most extensively used steels for drop hammers, and forging machine dies when production is large
D	Chromium-molybdenum vanadium steel	
E	Chromium-molybdenum steel	Dies for forging machines when production is large
F	Nickel-chromium-molybdenum-vanadium steel	Dies for mass production on hammers and presses. Die inserts
G	Special steel for nitriding	Die inserts for hammers and presses, surface hardened by nitriding after complete machining and heat treatment
H	Tungsten-chromium-vanadium steel	Small dies and die inserts for mass production on presses
I	Tungsten-chromium nickel-molybdenum and vanadium steel	Hot working dies and tools where resistance to wear and abrasion combined with toughness is necessary
J	Group of carbon steels	Cheap steels commonly used for trimming, blanking, piercing, and straightening, press tools for hot and cold working operations
K	Manganese steel	Tools of intricate shape for trimming, piercing, straightening, coining, etc. This steel is of non-shrinking type
L	Chromium-vanadium steel	Typical press tools for large production. The alloying element improves property and makes steel very suitable for cold working tools
M	Tungsten-chromium-vanadium steel	Tools for flash trimming and piercing and for hot and cold working operation. Specially suitable for mass production

### Heat Treatment of the Die Blocks

Heat treatment of die blocks is carried out before or after complete machining of the die. The machining of the die impression cannot be successfully accomplished when hardness exceeds 420 B.H.N. Carbon steel die blocks which can be only hardened to limited depths are often heat treated after preliminary machining but before final finish of the impression.

### Hardness of the Die Blocks

Hardness of the die blocks depends on the size and shape of the forging and varies between 280-500 B.H.N. Die blocks and die inserts of 460-500 B.H.N. are used only occasionally for simple, small and thin (4-6 mm.) forgings. The bulk of the die blocks used most commonly have hardness within the limits 280-450 B.H.N.

Nitrided inserts (see steel G, Table 1) are made usually with case 0.5-0.8 mm. thick. The hardness of the case is exceptionally high and it is reaching up to 800 B.H.N. when the core hardness usually never exceeds 450 B.H.N. Nitrided inserts are suitable only for the production of comparatively simple and small forgings.

The most popular steel for drop forging dies is steel C specified in Table 1. Die blocks made from nickel-chromium-molybdenum steel are usually supplied for machining, fully heat treated in four grades of hardness (Table 3).

**TABLE 3—HARDNESS OF DIE BLOCKS MADE FROM NICKEL-CHROMIUM-MOLYBDENUM STEEL**  
(See specification Steel C, Table 1)

Brinell Hardness No. D = 10 mm. P = 3000 kg.	Diam. of the ball impression in mm.	Uses
420-395	3.00-3.10	Dies with shallow impression for small and simple forgings. Depths of the impression up to 10 mm.
390-365	3.10-3.20	Most widely used dies for medium size forgings. Depths of the impression up to 30 mm.
360-365	3.20-3.35	Dies for large and intricate shape forgings. Depths of the impression up to 80 mm.
320-300	3.35-3.50	Dies with very deep and intricate impressions

### Heat Treatment of Other Tools

Heat treatment of other tools used in the forge for flash trimming, piercing, straightening and coining depends on the type of steel and duty performed by the tool. Trimming blades are hardened and tempered to 350-600 B.H.N. The cold working tools require a minimum hardness of 500 B.H.N. Punches of the flash trimming tools which only push forging through trimming blades are heat treated to 270-320 B.H.N. When production is small they may be used without heat treatment. Piercing punches which must possess sharp and hard cutting edges are always heat treated to 400-600 B.H.N. Tools and dies for straightening and coining are usually used in heat treated condition. The final hardness depends on the nature of work and generally it varies between 350-550 B.H.N.

## DROP FORGING DIES

### Fastening of the Die

Fastening of the dies to the hammer anvil block is made by dove-tail and long tapered wedges called "keys" or by large diameter screws passing

through heavy forged steel blocks known as "poppetts" secured to the block by steel cotters. The top die of the pair of the dies is fastened directly to the tup by the key driven along the dove-tailed shunk of the die. The low die is fixed in a similar manner usually not directly to the base of the hammer but to the large steel block called the die-holder. The die-holder itself is fixed to the anvil either by dove-tail and key or by 4 (6 for large hammers) poppet screws (Fig. 2). In both, tup and die-holder, the dies are secured against the movement from front to back by dowels closely fitted into recesses provided in tup and die-holder.

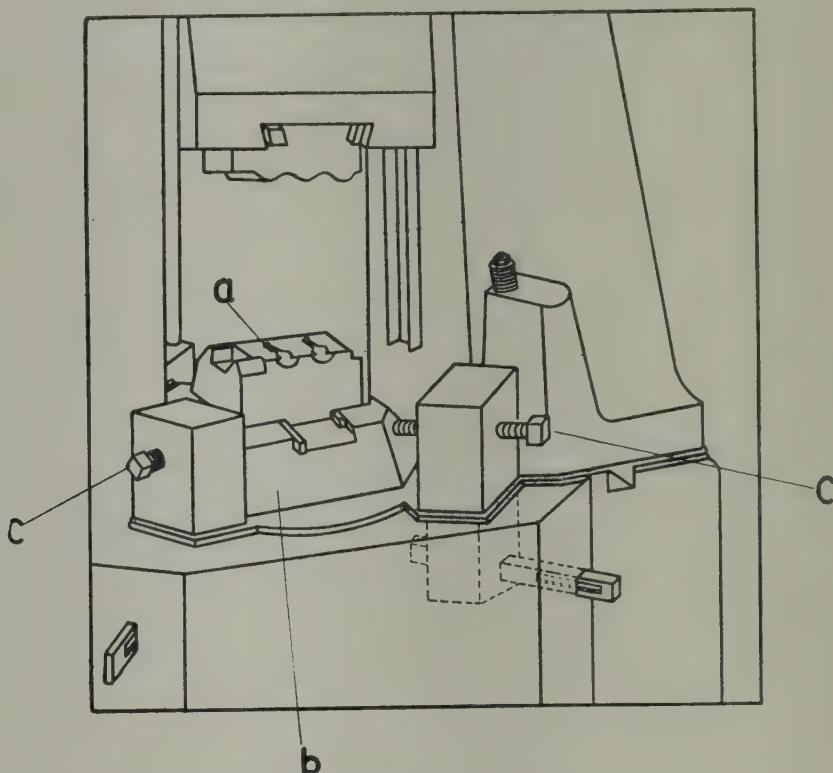


FIG. 2—FASTENING OF THE DIE TO THE ANVIL BLOCK OF THE HAMMER BY DIE HOLDER AND FOUR POPPET SCREWS: (a) Lower Die-holder; (b) Anvil Block; (c) Poppett screws

Fastening by poppet screws simplifies the setting and correct matching of the die but poppet screws have a tendency to loosen and it is necessary to tighten them up several times during the day. Fastening of the lower die by poppet screws directly to the base of the hammer without the die-holder is used very seldom and only for big hammers.

The dove-tails of the dies together with long taper keys prevent side movement of the dies and the possibility of longitudinal movement is eliminated by cross keys. Figure 3a illustrates the method of fastening the top and lower die by two long keys "k" and by centrally situated cross-key "i". The standard dimensions of these keys for various sizes of drop hammers are given in Table 4. For large hammers above 3 ton capacity and in cases of long die blocks two or even three cross keys are used.

The other fastening principles of die commonly used in U.S.A. are illustrated in Fig. 3b. Here there is only one large key “m” and one cross key “n” situated on the edge of the die dove-tail. As to the usefulness of these two methods opinions are divided.

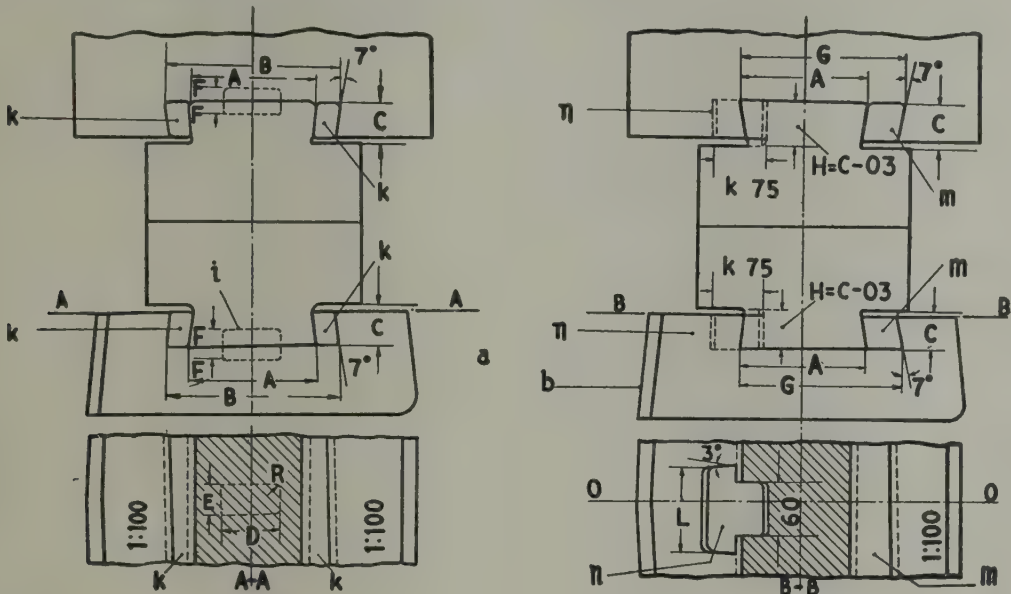


FIG. 3—FASTENING OF THE DIE TO THE DIE-HOLDER OF THE HAMMER: (a) Fastening by two keys and centrally situated one cross key ; (b) Fastening by key and one cross key situated on the edge of the die dove-tail

TABLE 4—STANDARD DIMENSION OF THE DIE FASTENING ELEMENTS (in mm.) (See also Fig. 3)

Size of the Hammer	250 kg.	350 kg.	500 kg.	750 kg.	1t	1.5t	2t	2.5t	3t	4t	5t	6t
A	120	150	150	200	200	200	270	270	270	330	330	330
B	185	215	215	270	270	270	360	360	360	435	435	435
C	50	50	50	50	50	50	65	65	65	75	75	75
D	60	60	60	80	80	80	100	100	100	120	120	120
E	40	40	40	50	50	50	50	50	50	60	60	60
F	20	20	20	25	25	25	25	25	25	30	30	30
R	12.5	12.5	12.5	15	15	15	15	15	15	20	20	20
G	170	200	200	250	250	250	330	330	330	390	390	390
K	50	50	50	50	50	50	65	65	65	65	65	65
L	75	75	75	75	75	75	90	90	90	90	90	90

Classification of the Drop Forging Dies

Drop forging dies may be classified into two groups:

- SINGLE IMPRESSION DIE
- Containing only one impression which is the finishing impression
- MULTI-IMPRESSON DIE
- Containing finishing impression and one or more auxiliary impressions for preliminary forging operations

Impressions of the Die (Fig. 4)

Impressions of the die are machined on the face of the block and this

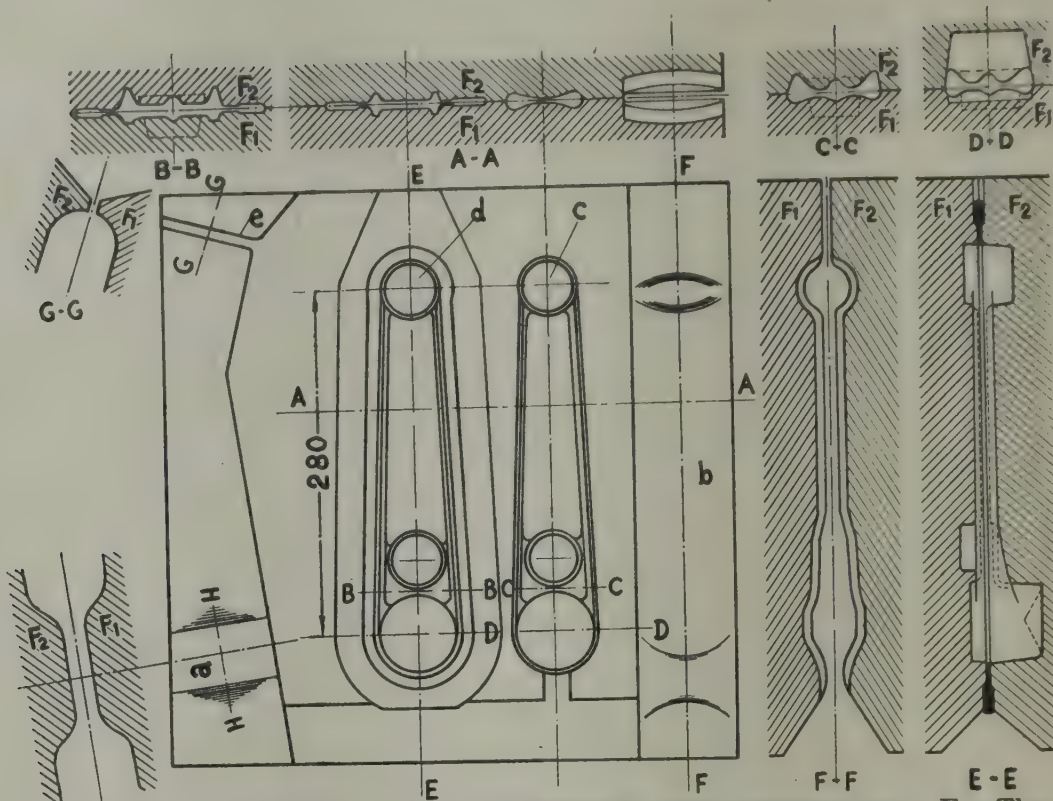


FIG. 4—TYPICAL EXAMPLE OF MULTI-IMPRESSION DROP FORGING DIE:  $F_1$ —The Lower Die;  $F_2$ —The Top Die. (In all drawings illustrating drop forging dies, the plan represents the lower die): (a) Fuller, (b) Edger, (c) Blocking impression, (d) Finishing impression and (e) Cut-off

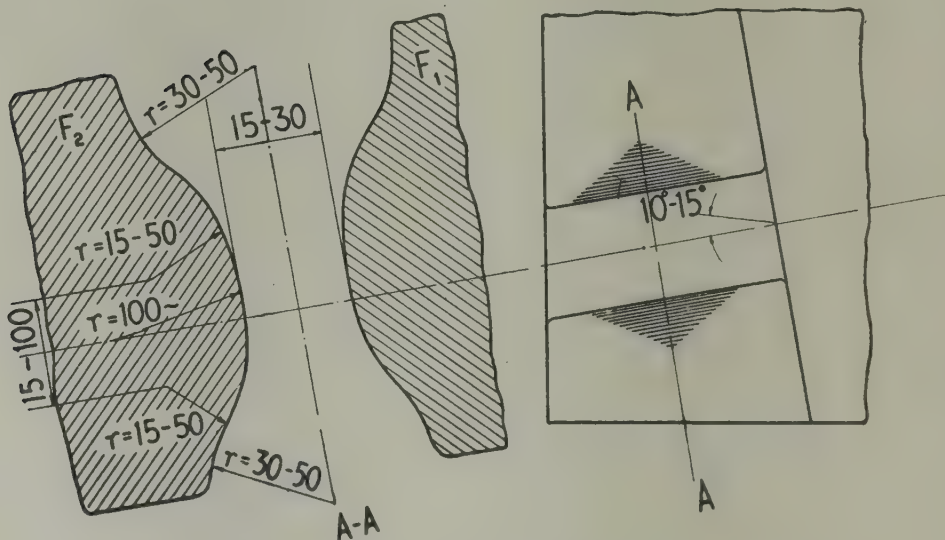


FIG. 5—STANDARD DIMENSION OF THE FULLERING IMPRESSION FOR BARS OF CROSS SECTION VARYING FROM  $20 \times 20$  mm. UP TO  $60 \times 60$  mm. ; INDICATED DIMENSIONS ARE ONLY FOR GENERAL GUIDANCE OF THE DIE DESIGNER. DEPTHS OF IMPRESSIONS SHOULD BE GRADUALLY INCREASED FOR BIGGER CROSS SECTIONS OF THE BARS AND ITS LENGTHS SHOULD BE IN PROPORTION WITH LENGTHS OF THE FORGING:  $F_1$ —The Lower Die;  $F_2$ —The Top Die

operation may be performed on one or more of standard or special machine tools such as the lathe, boring machine, milling machine, etc.

The finishing touch to the impression is given by hand work such as scraping, filing, grinding and polishing. In the multi-impression dies together with finishing impression, there are several types of auxiliary impressions required on the set of the dies such as fuller, edger, bender, flattening impression and blocking impression.

### Fuller (Fig. 4a and Fig. 5)

The primary function of the fullering impression is to reduce the cross sectional area of the forging stock. The bar is turned 90° between each blow.

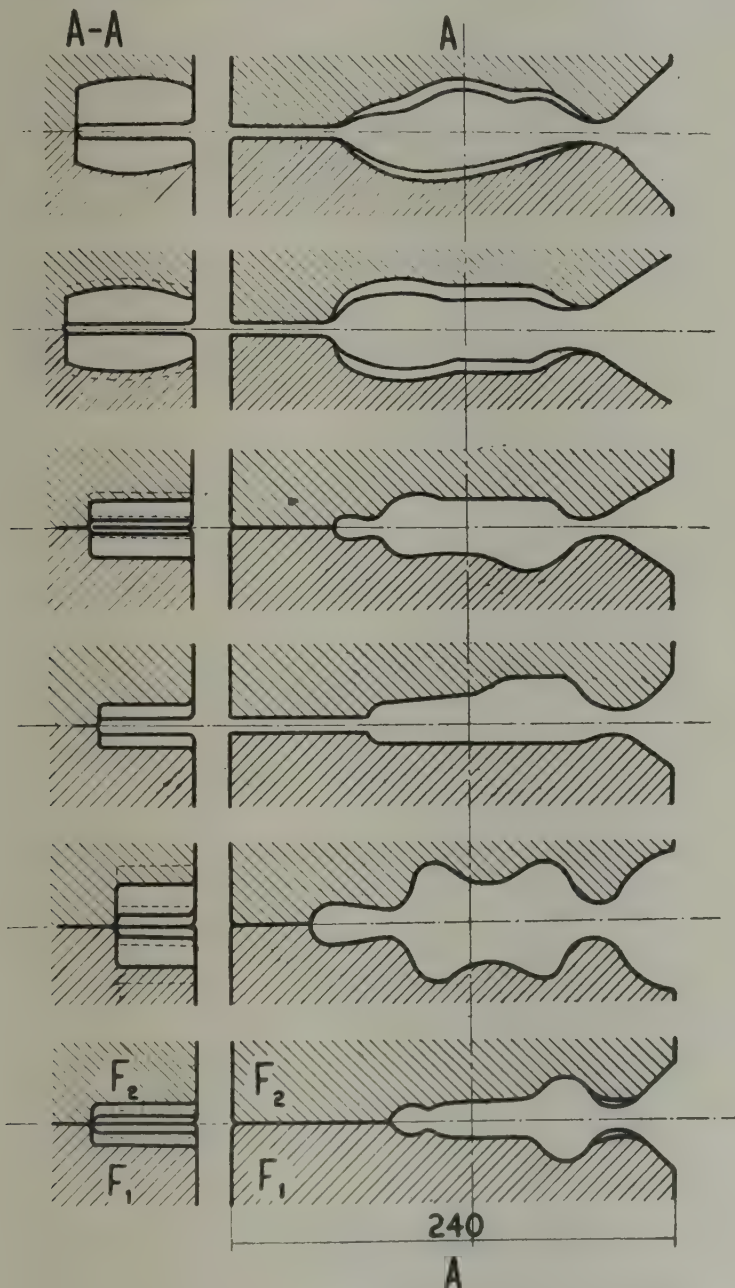


FIG. 6—TYPICAL EXAMPLES OF EDGING IMPRESSIONS:  $F_1$ —The Lower Die;  $F_2$ —The Top Die

**Edger (Fig. 4b and Fig. 6)**

The general contour of the edging impression corresponds to the shape of the final forging. This impression which is open partly or completely on the sides allows the metal to flow in a side way direction but confines it in the front and back ends. The edging impressions move some of the metal from one portion of the forging stock to another part so as to distribute the metal in relation to the finish forging. The edger may be symmetrical or non-symmetrical to the partition plan of the die. When the non-symmetrical edging impression is used the bar is turned  $90^\circ$  between each blow but it is essential to turn it once in the left and then in the right direction. However, for symmetrical edgers that is not necessary.

**Bender (Fig. 7)**

Bending impression is used for bending the stock after it has been edged or fullered and edged. The shape of the bending impression is related to the contour of the forging so that it can position the stock into a suitable shape for the finishing impression. The bending is as a rule a one-blow operation.

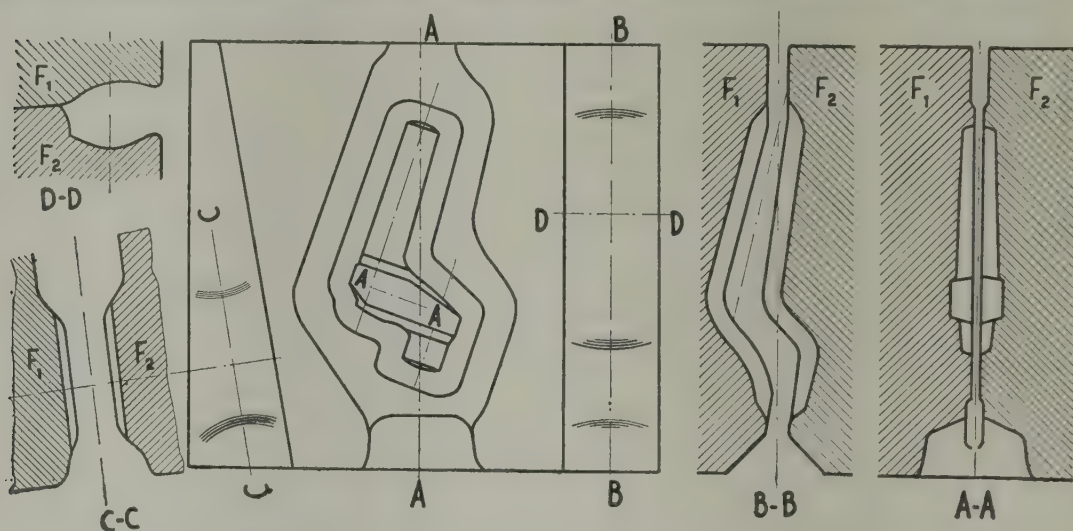


FIG. 7—TYPICAL EXAMPLE OF MULTI-IMPRESSION DIE WITH FULLERING AND BENDING IMPRESSIONS:  $F_1$ —The Lower Die;  $F_2$ —The Top Die

**Flattening Impression (Fig. 8)**

Sometimes there is need for flattening the stock before passing it to the final impression. This is done in the flattening impression, usually very simple and situated in one of the front corners of the die block.

**Blocking Impression (Fig. 4c)**

The blocking impression has a shape of the finishing impression but with stream lined corners, holes, and intricate details. It is added to reduce the wear of the finishing impression and it eliminates all forging defects such as laps and cold shuts.

**Cut-off (Fig. 4e, Figs. 8 and 9)**

When forgings are made from the bar they must be cut off after the drop forging operation is finished. This is done either by a special side cutter of

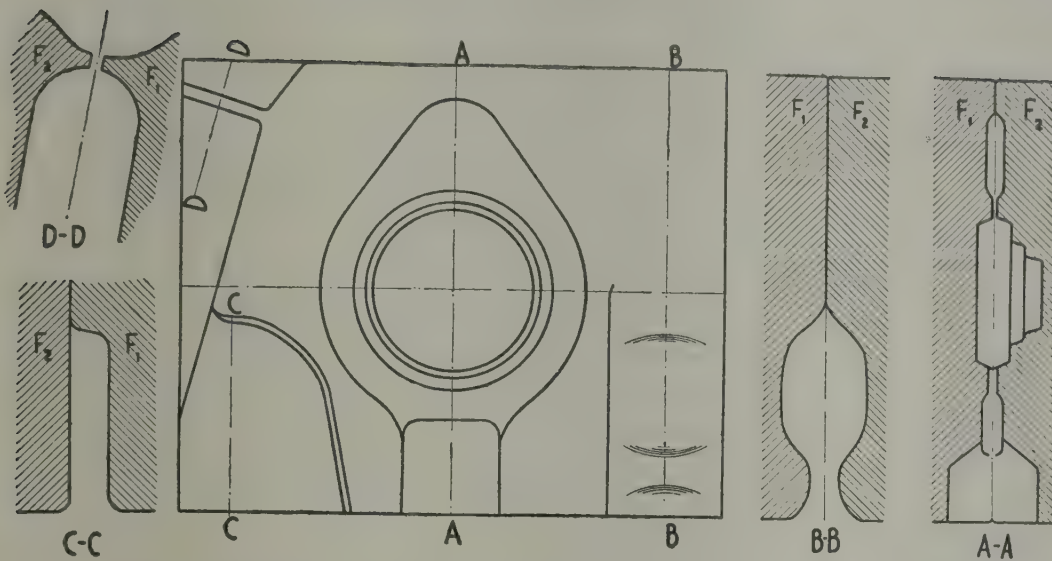


FIG. 8—TYPICAL EXAMPLE OF MULTI-IMPRESSION DIE WITH EDGING, FLATTENING AND CUT-OFF IMPRESSIONS:  $F_1$ —The Lower Die ;  $F_2$ —The Top Die

the trimming press or by the cut-off impression milled usually in the left back corner of the die block.

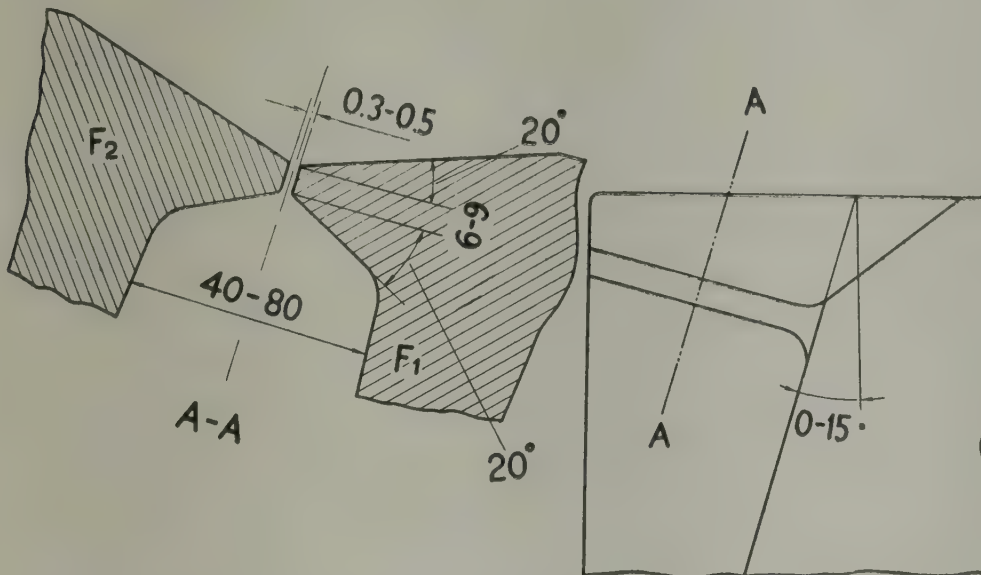


FIG. 9—STANDARD DIMENSIONS OF THE CUT-OFF IMPRESSION FOR THE BARS OF CROSS SECTION VARYING FROM  $20 \times 20$  UP TO  $60 \times 60$  mm. INDICATED DIMENSIONS ARE ONLY FOR GENERAL GUIDANCE OF THE DIE DESIGNER ; DEPTHS OF IMPRESSIONS SHOULD BE GRADUALLY INCREASED FOR BIGGER CROSS SECTION OF THE BARS:  $F_1$ —The Lower Die ;  $F_2$ —The Top Die

### The Finishing Impression (Fig. 4d and Fig. 10)

The finishing impression exactly corresponds to the final shape of the forging and is located in the middle of the block, but it is not necessarily in its central axle. However, it is vital to locate the final impression in such a manner that there will be no horizontal forces which give a side truss and may cause the die to shift. Generally speaking the imaginary centre of gravity of the forging counter should be on the central axle of the hammer top.

### Shrinkage Allowances

Shrinkage allowances used by the tool room when making die impressions (contraction rule) are usually applied for steel forgings in the following manner:

1. Forgings which after flash trimming are not straightened by a light blow in the final impression of the die— $1/65$ th of the nominal dimension.
2. Forgings which after flash trimming are returned to final impression of the die for straightening by light blow— $1/100$ th of the nominal dimension.
3. For longitudinal dimension in long and thin forgings— $1/65$ th of the nominal dimension.

For other metals the shrinkage allowances are fixed according to the co-efficient of expansion of metal and temperature in which the forging operation is finished.

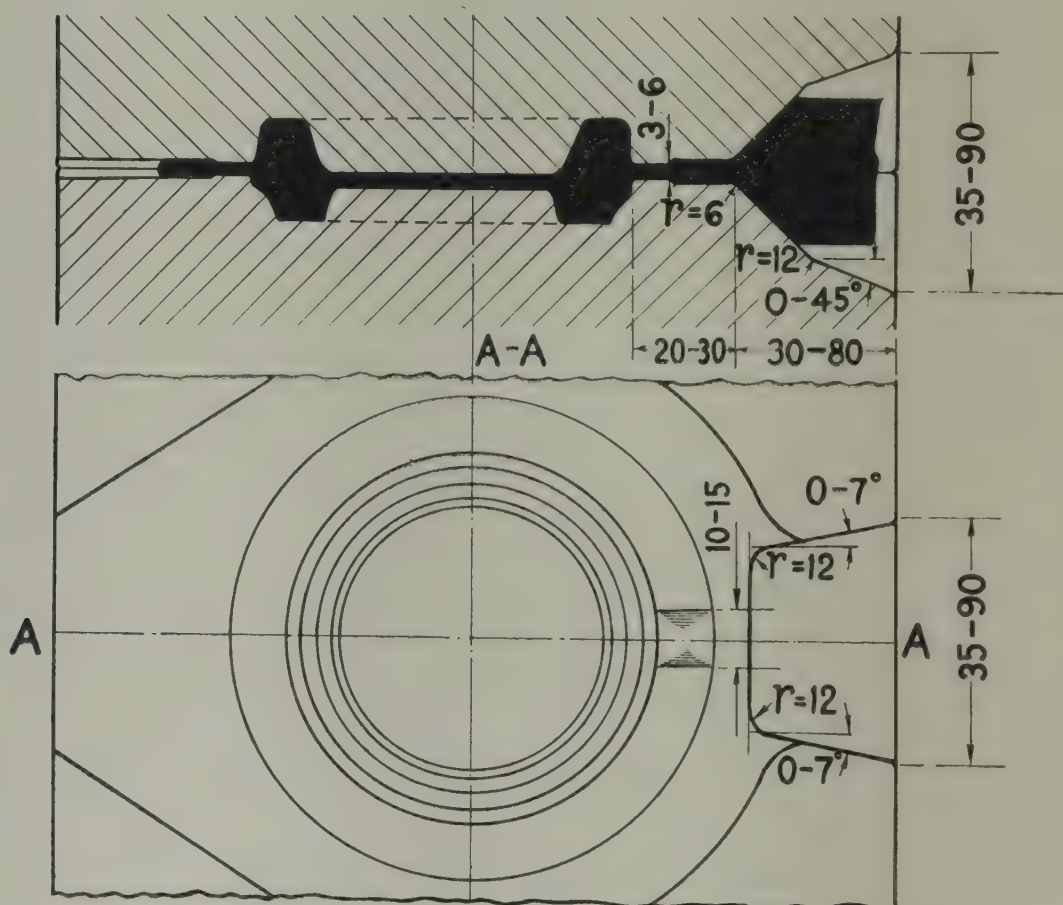


FIG. 10—TYPICAL STANDARD FINISHING IMPRESSION OF THE DROP FORGING DIES FOR THE BARS OF CROSS SECTION VARYING FROM  $20 \times 20$  UP TO  $60 \times 60$  mm. INDICATED DIMENSIONS ARE ONLY FOR GENERAL GUIDANCE OF THE DIE DESIGNER. DIMENSION SHOULD BE GRADUALLY INCREASED FOR THE BIGGER CROSS SECTION OF THE BARS.

### Flash Gutter (Fig. 11)

The flow of plastic metal under the blows of the drop hammer proceeds first to fill up the impression, then the small quantity of metal moves into the shallow cavity provided around the finishing impression of the die. These

small cavities which are directly outside the die impression are known as the flash gutter. The flash gutter is separated from the impression of the die by a narrow passage the dimension of which must be very carefully considered by the die designers. Standard dimensions of the flash gutter are given in Table 5. In the same table other dimensions concerning the finish shape of the impression are also given.

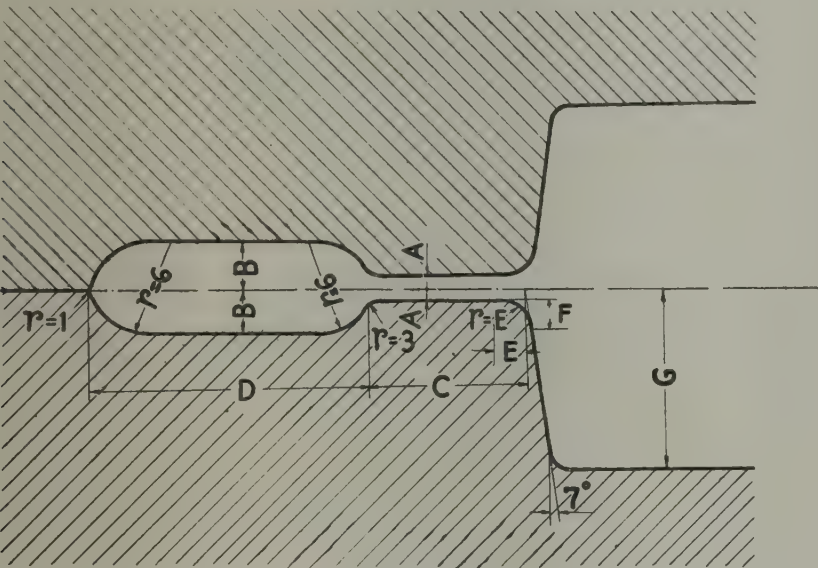


FIG. 11—STANDARD DIMENSION OF THE FLASH GUTTER  
(See also Table 5)

TABLE 5—STANDARD DIMENSION OF THE FLASH GUTTER  
(See also Fig. 11)

Dimension	Size of the hammer										
	250	350	500	750	1t	1.5t	2t	2.5t	3t	3.5t	4t
2A from	0.8	1.0	1.1	1.2	1.4	1.5	2.0	2.2	2.5	2.7	3.0
up to	1.2	1.3	1.7	2.0	2.3	3.0	3.5	4.0	4.5	3.0	5.5
B mm.	3	3	3	3	3	3	4	4	4	5	5
C mm.	8	8	10	10	12	12	14	14	15	15	16
D mm.	15	15	20	20	25	25	30	30	35	35	40

TABLE 6—STANDARD DIMENSION OF FINISHING IMPRESSION  
(See also Fig. 11)

Dimension G, mm.	5	10	20	30	40	50	60	70	80	90	100
E mm.	1.0	2.0	2.3	2.5	2.8	3.1	3.4	3.7	4.0	4.3	4.6
F mm.	2	3	5	8	10	13	15	18	20	23	25

Die Parting Surface (Fig. 12)

Die parting surface divides the final impression into the two parts from which one is made in the top die and the other in the lower die. The location of the parting surface is very important and has considerable influence on the flow of metal. When deciding upon this fundamental factor the die designer should

always remember that hot metal fills up the top die first and that the deep and intricate part of the impression should be cut in the top die. When, owing to the shape of forging, a complete impression is arrived at in one part of the die, this should be the top die. In such a case, the lower die remains without the impression and will have only locating elements.

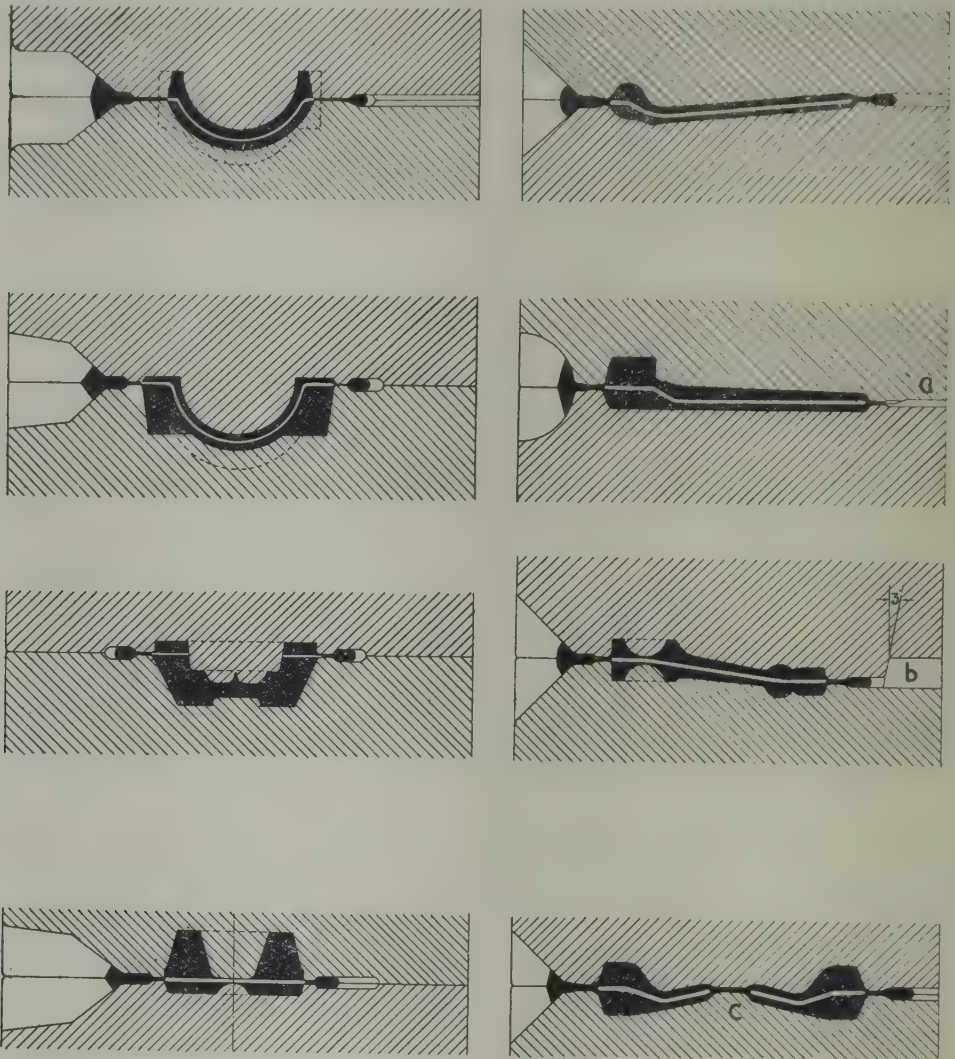


FIG. 12—TYPICAL EXAMPLES OF DIE PARTING SURFACES IN DROP FORGING DIES: (a,b)—Counter-inclined ; (c)—Locked die balance by inclusion of two final impressions opposite to each other

### Counter-inclined Planes (Figs. 13a and b)

To balance the die against the side truss caused by the inclined plane of the die parting surface which may throw the top and bottom die out of register a counter-inclined plane must be added.

### Locked Dies

For balancing side truss and sometimes only for quick setting of the intricate impression die on the hammer it is desirable to use the locked die of the type illustrated in Fig. 13.

### Locating Elements

When complete impression is made only in one part of the die (top die) or if impression in the lower die is less than 3 mm. the die designer must include small oval shape locating elements shown on Fig. 13. This drawing illustrates also the use of the locating elements when the shape of the round forging does not allow it to be turned around in the die impression.

Different types of locating elements usually of the conical shape (Fig. 13) are used to secure the proper location of the forging on the trimming tool. The flash trimming operation is usually done in the reverse position to that in which forging is made in the die and for this reason locating recesses are provided in the top die.

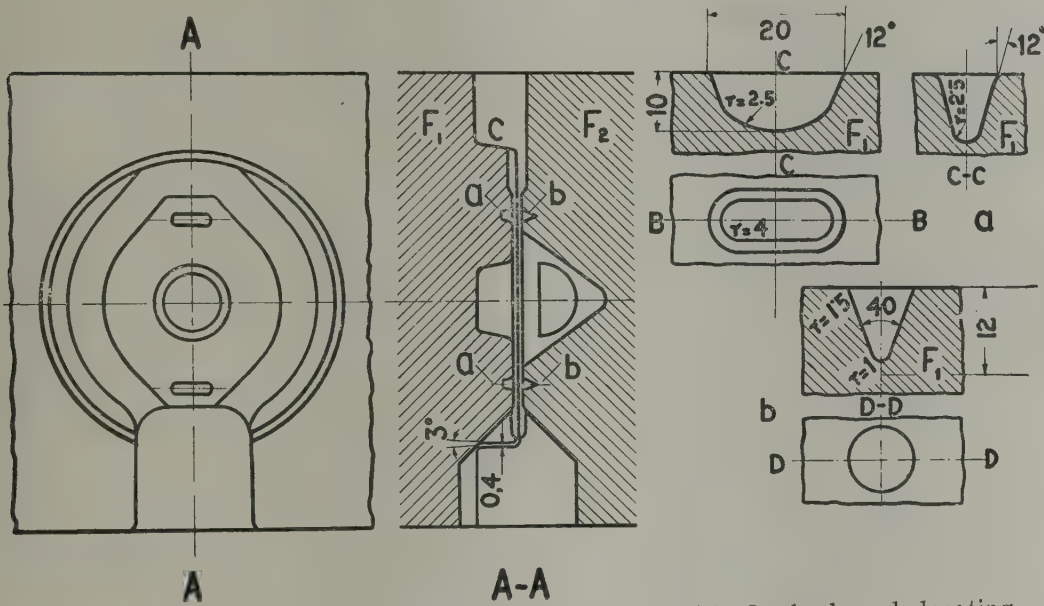


FIG. 13—TYPICAL STANDARD LOCATED ELEMENT: (a) Oval shaped locating element to locate part during the forging operation; (b) Conical shaped locating element to locate part during flash trimming operation; (c) Die locating impression

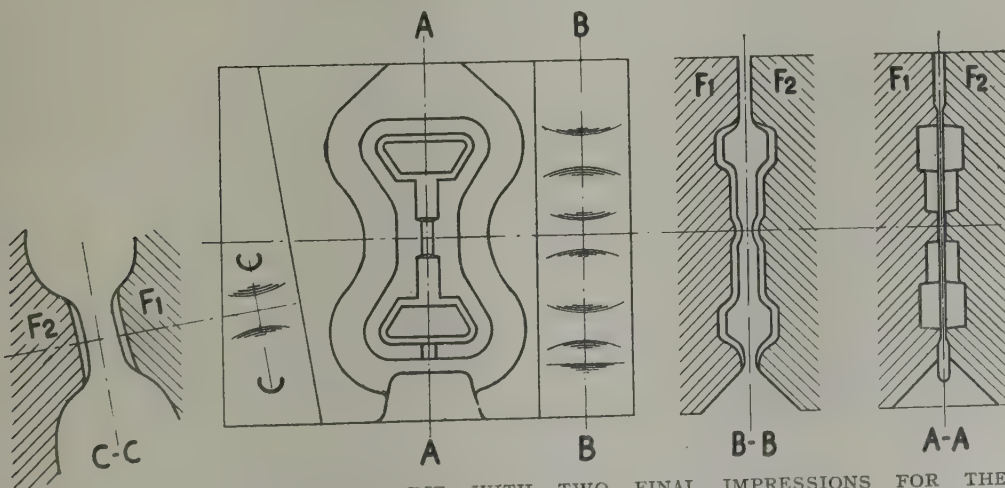


FIG. 14—MULTI-IMPRESSON DIE WITH TWO FINAL IMPRESSIONS FOR THE PRODUCTION OF TWO FORGINGS AT A TIME

## SOME EXAMPLES OF DROP FORGING DIE DESIGN

The modern technique of drop forging die design based on the principles previously described may be well illustrated by a few typical drawings. To

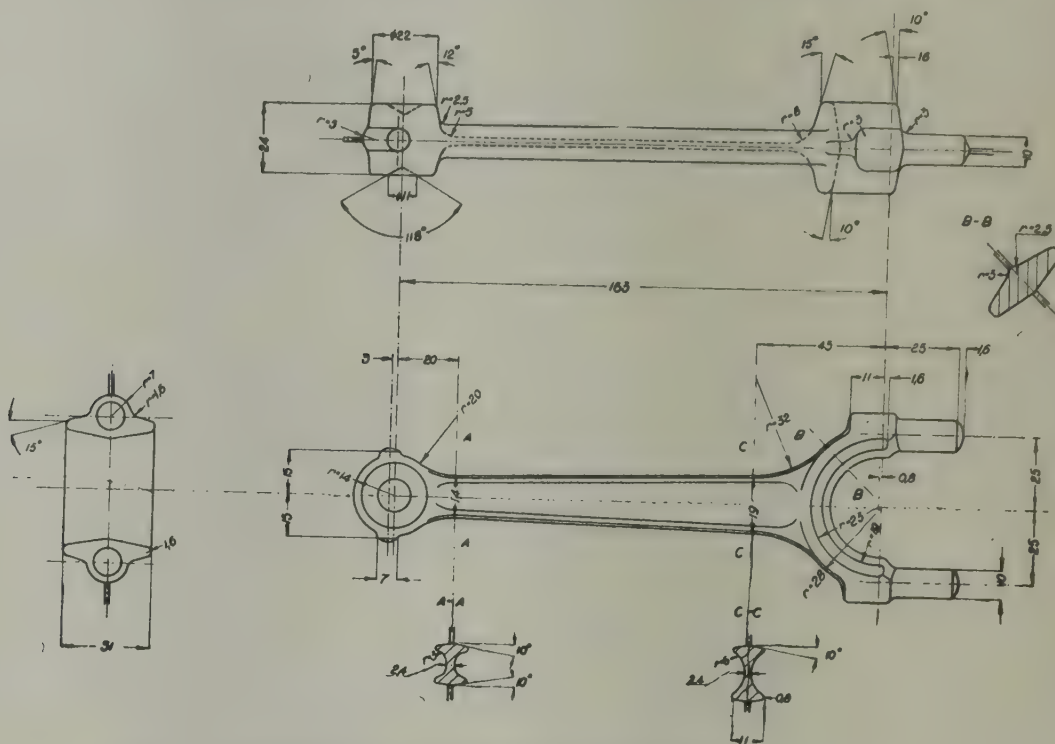


FIG. 15—DRAWING OF CONNECTING ROD FOR FORD MOTOR CAR

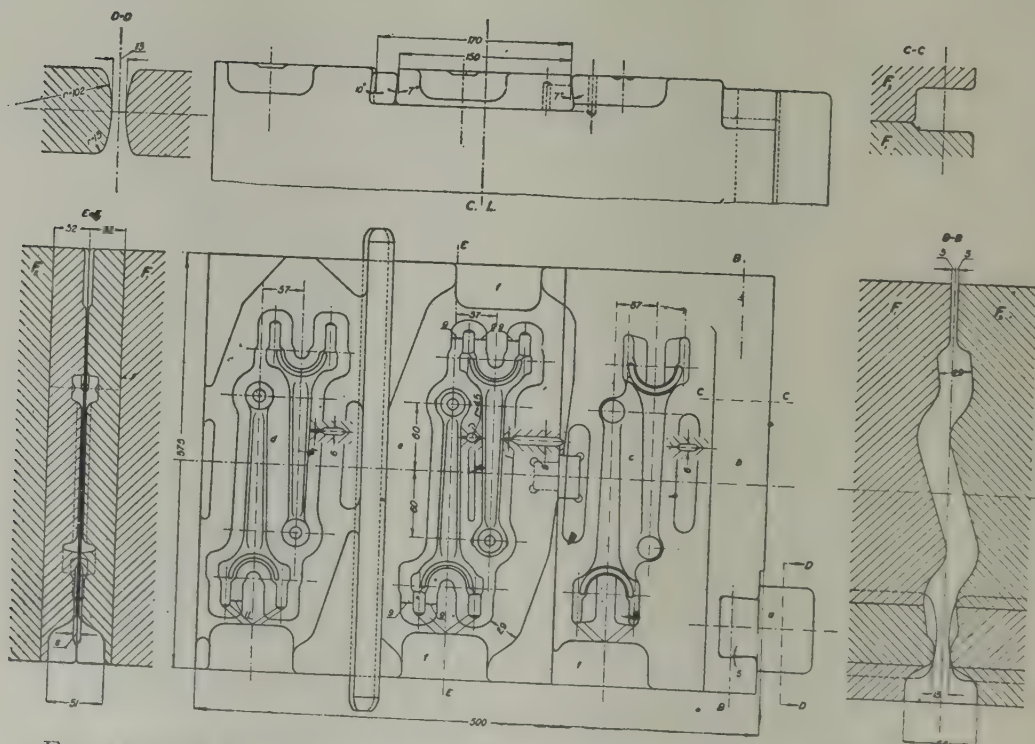


FIG 16—DIE DESIGN FOR PRODUCTION OF FORD MOTOR CAR CONNECTING ROD

illustrate the differences which may appear in solving this problem a typical part of internal combustion engine, *viz.*, connecting rod is chosen as a main example.

Figs. 15, 17 and 19 contain drawings of similar sizes of connecting rods

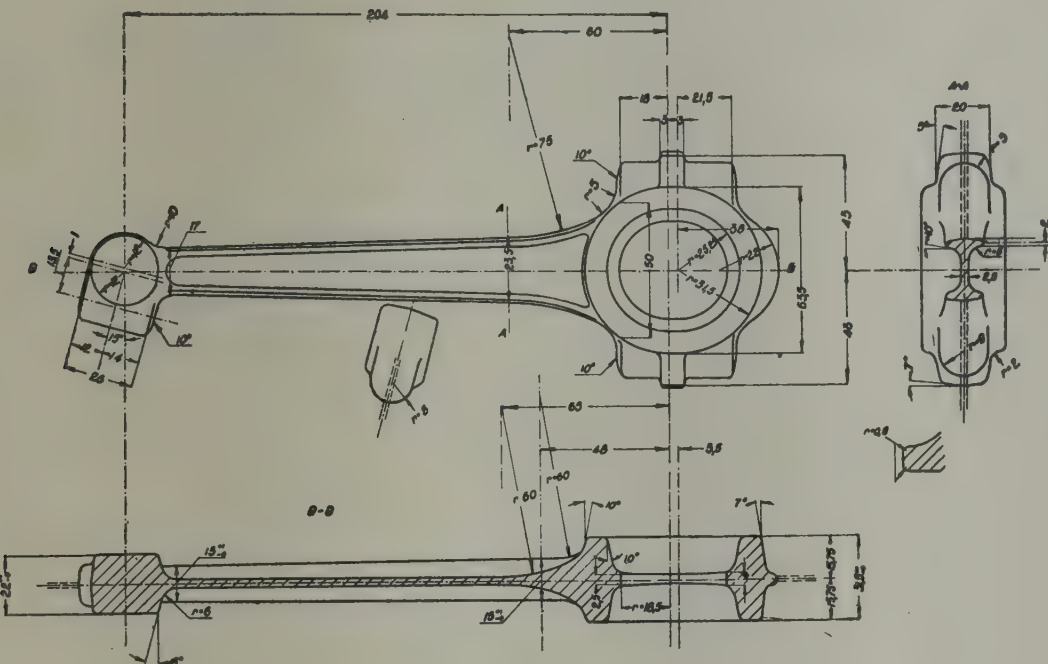


FIG. 17—DRAWING OF CONNECTING ROD FOR MORRIS MOTOR CAR

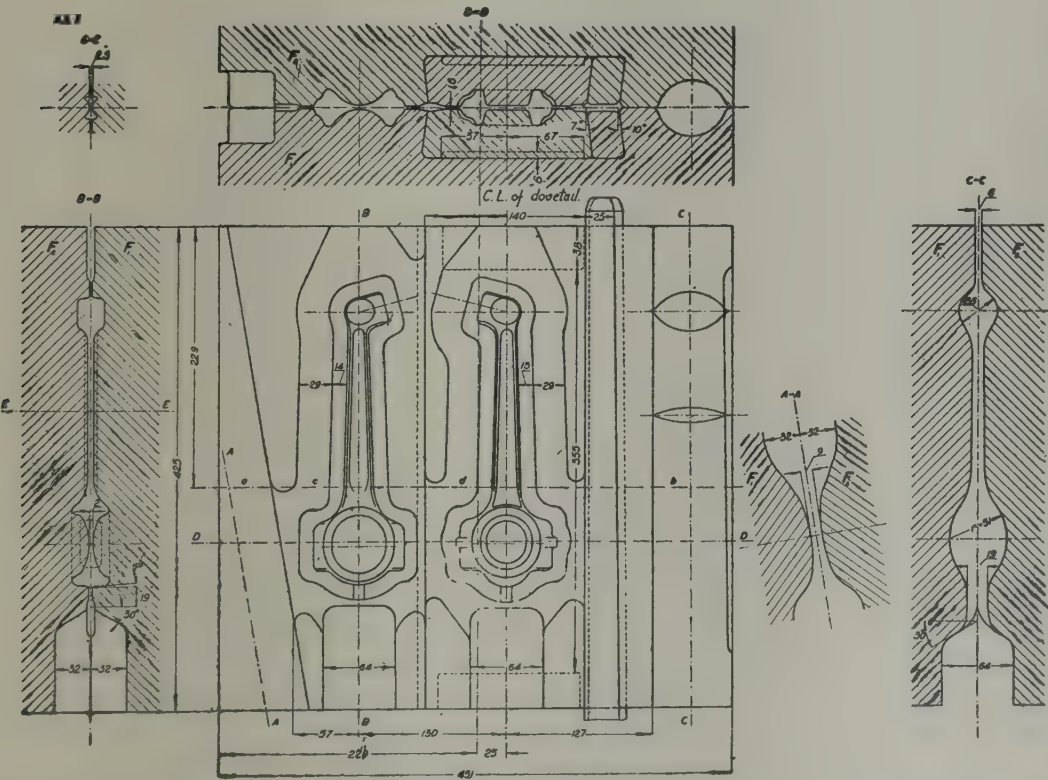


FIG. 18—DIE DESIGN FOR PRODUCTION OF MORRIS MOTOR CAR CONNECTING ROD



The connecting rod (Fig. 19) is made on two hammer equipped with dies (Figs. 20 and 21) from which the die illustrated on Fig. 20 has only a preliminary impression. The work is done on two separate hammers installed side by side so that the whole process can be made from the same heat.

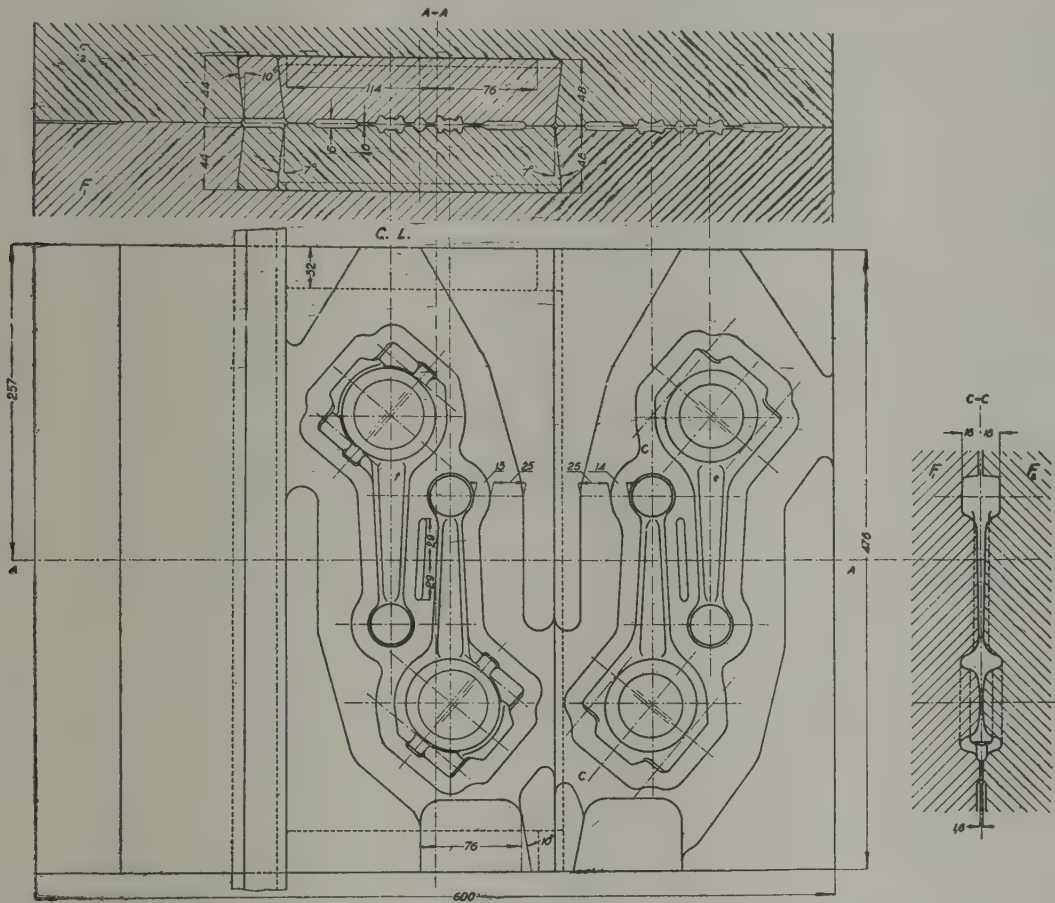


FIG. 21--DIE WITH FINAL IMPRESSION DESIGN FOR PRODUCTION OF STANDARD  
MOTOR CAR CONNECTING ROD

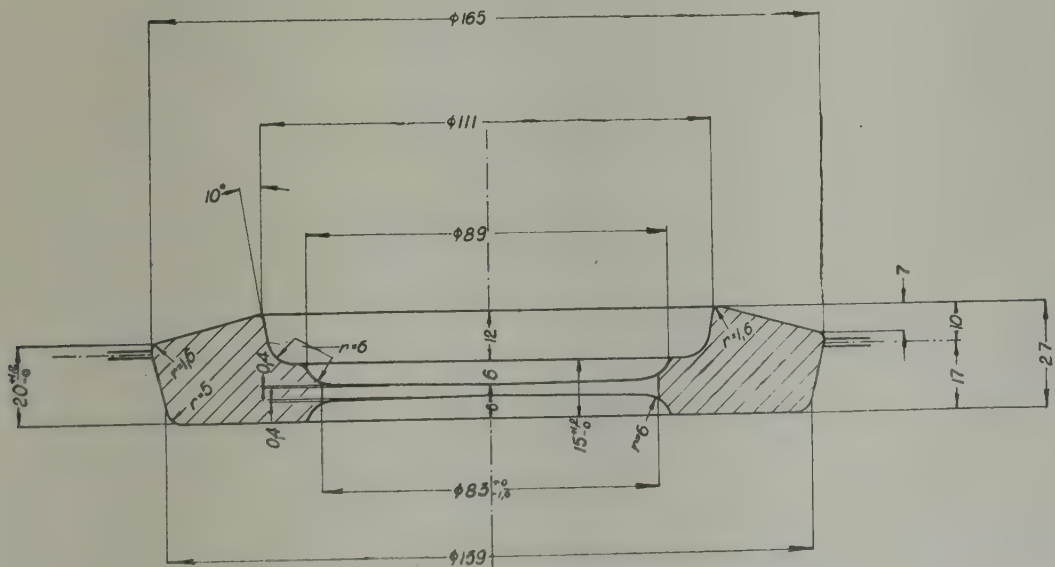


FIG. 22—DRAWING OF GEAR BLANK FOR MOTOR CAR

The drawings (Figs. 22 and 23) illustrate the gear blank and the die design for its production.

In the design of the dies for these connecting rods the exchangeable inserts are adopted which considerably prolong the life of the dies, and simplify their maintenance.

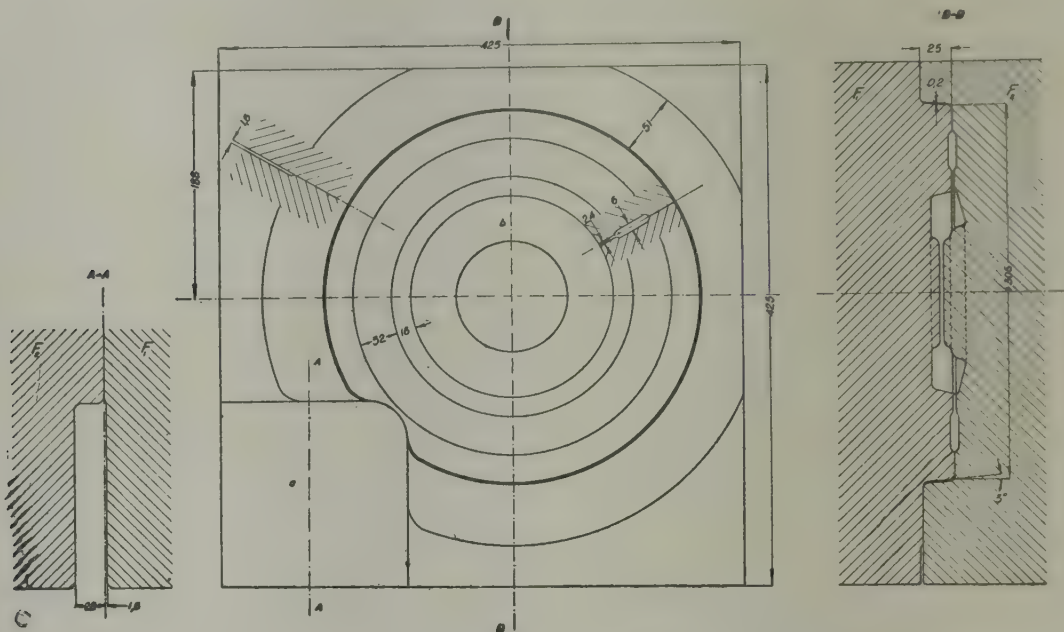


FIG. 23—DIE DESIGN FOR PRODUCTION OF MOTOR CAR GEAR BLANK

### DIES FOR FORGING MACHINE

The sequence of making a forging on the forging machine comprises one to five consecutive upsetting and forming operations. Die design for forging machine is based on three practical rules governing the action of metal flow in the die of forging machine.

**RULE NO. 1.** The lengths of unsupported bar that can be upset in one blow without buckling must not be more than three times the diameter of the bar (Fig. 24).

In practice it is better that the length of the unsupported stock is within  $2\frac{1}{2}$  times the diameter.

**RULE NO. 2.** Lengths of the bar more than three times its diameter can be successfully upset in one blow provided the diameter of the upset made is not more than  $1\frac{1}{2}$  times the diameter of the bar (Fig. 25).

It is advisable in practice not to exceed 1.3 times the diameter of the bar. Analysing this rule it is quite clear that when upsetting the lengths of the bar exceeding 3 times the diameter, the stock will at once begin to buckle but on account of the limited diameter of the die impression the stock is immediately supported on the sides of the die which eliminate further buckling. However, if part of the bar is outside the die impression it is necessary to remember the third rule which is actually a combination of the first and second rule.

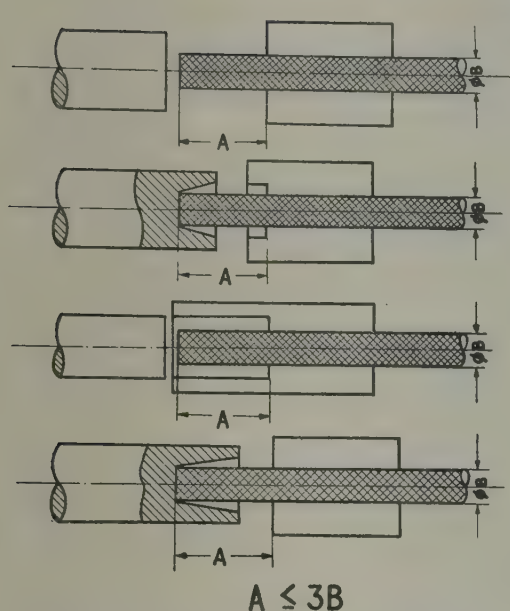


FIG. 24—FOUR DIFFERENT EXAMPLES OF RULE NO. 1 GOVERNING UPSETTING PROCESS

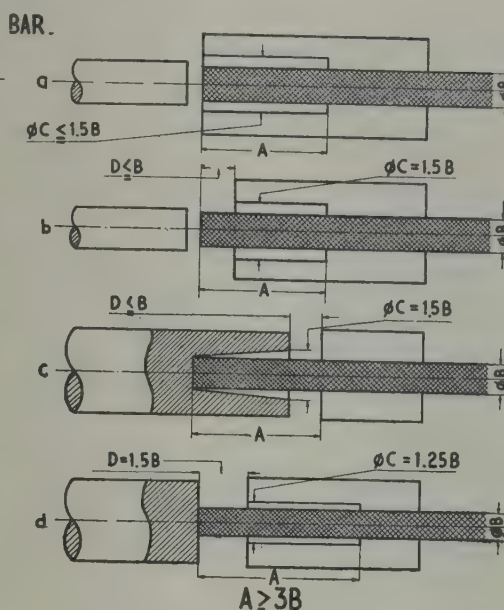


FIG. 25—FOUR DIFFERENT EXAMPLES OF RULES NOS. 2 & 3 GOVERNING UPSETTING PROCESS

**RULE No. 3.** Upsetting the lengths exceeding 3 times the diameter of the bar when the diameter of the upset is  $1\frac{1}{2}$  times the diameter of the bar, the amount of unsupported stock beyond the face of the die must not exceed one diameter of the bar (Fig. 25).

By reducing the diameter of the die impression it is possible to increase unsupported stock lengths. For example if the diameter of the die impression is only 1.25 times the diameter of the bar the supported lengths may be 1.5 times the bar diameter. On decreasing further the diameter of the die impression, the lengths of unsupported stock may be increased accordingly and finally the condition specified for the first rule may be obtained.

### Classification of Dies for Forging Machines

Dies for forging machines may be classified in the following way: (1) Dies for upsetting and forming ends of the bar (Fig. 26); (2) dies for producing small forgings which after upsetting and forming will be cut off from the end of the bar (Fig. 27); (3) sliding dies for upsetting the bar in two or more places at the same stroke of the machine (Fig. 28); (4) deep piercing dies; and (5) special dies for aiding drop hammer or forging presses.

### Die Blocks for Forging Machines

Die blocks for forging machines are made from heat treated steels similar to those used for drop forging dies. They may be built up from small blocks (Fig. 26 and Fig. 27B) each block with a single impression or they may be produced from single large blocks (Fig. 27A).

In the build-up type die repairs of worn out impression can be arranged simply by removing and replacing only one section of the die. This makes them most suitable for mass production when frequent repairs of the die are

necessary. Solid dies are cheaper in initial cost but in the long run repairs are more costly and difficult.

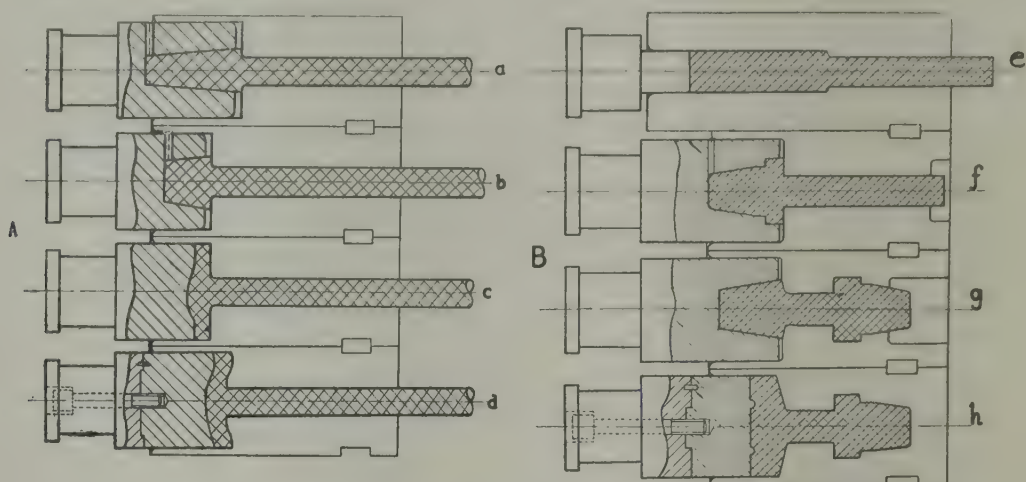


FIG. 26—TYPICAL EXAMPLES OF THE DIES FOR UPSETTING AND FORMING ENDS OF THE BARS: A—Die for production of rear axle for motor vehicle; B—Die for production of combined gear blank: (a) First upsetting; (b) Second upsetting; (c) Preliminary forming; (d) Final forming; (e) Upsetting of the first end; (f) Forming of the first end; (g) Upsetting of the second end; (h) Forming of the second end

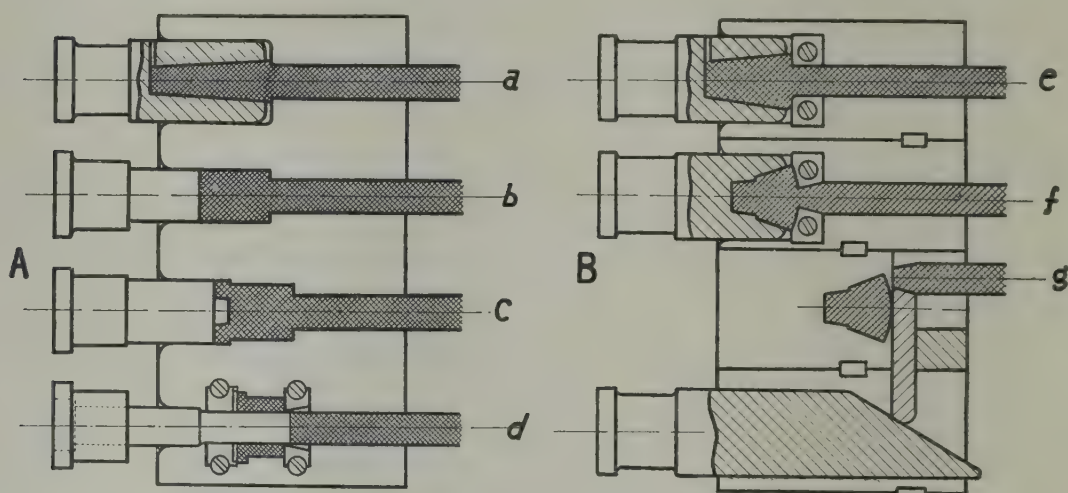


FIG. 27—TYPICAL EXAMPLES OF THE DIES FOR PRODUCING SMALL FORGING: A—Die for production of small bush; B—Die for production of bevel gear blank: (a) First upsetting; (b) Second upsetting; (c) Final forming; (d) Shearing off the forging from the bar; (e) First upsetting; (f) Final forming; (g) Cutting off the forging from the bar by vertical cutter

### Die Inserts

The parts of the die exposed to excessive wear caused by flow of metal are usually designed with small exchangeable inserts. These inserts are commonly used for punch faces (Fig. 26), piercing elements, necking section, etc. (Fig. 27).

### Die Impressions

When designing dies for forging machines special attention should be paid to the rules described already governing upsetting processes and to the fact

that the die impressions for the consecutive upsetting and forging operations must have the same volume. It must also be remembered that the taper-upset is the most desirable shape for upsetting the metal prior to the final forming. When the die impression is deep it is always very desirable to drill across the wall of the punch a small ventilating hole through which the air trapped underneath the metal may escape (Figs. 26 & 27).

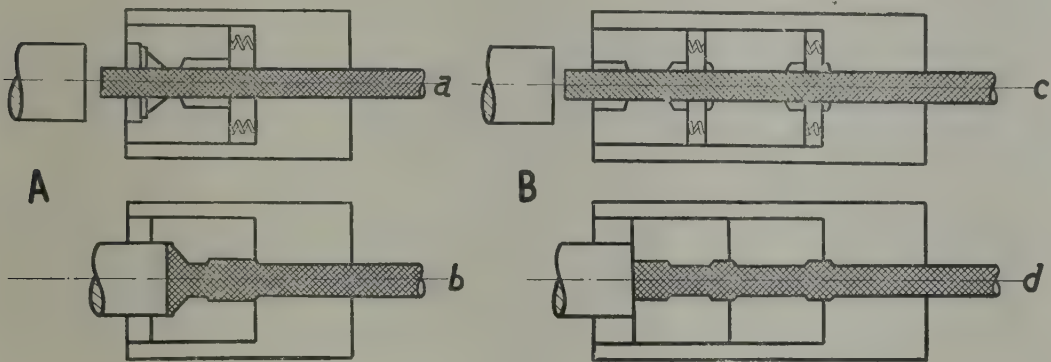


FIG. 28—DESIGN PRINCIPLES FOR SLIDING DIES: A—Die for upsetting the end of the bar in two places: (a) Die in open position; (b) Die in close position; B—Die for upsetting the bar in three places: (c) Die in open position; (d) Die in close position

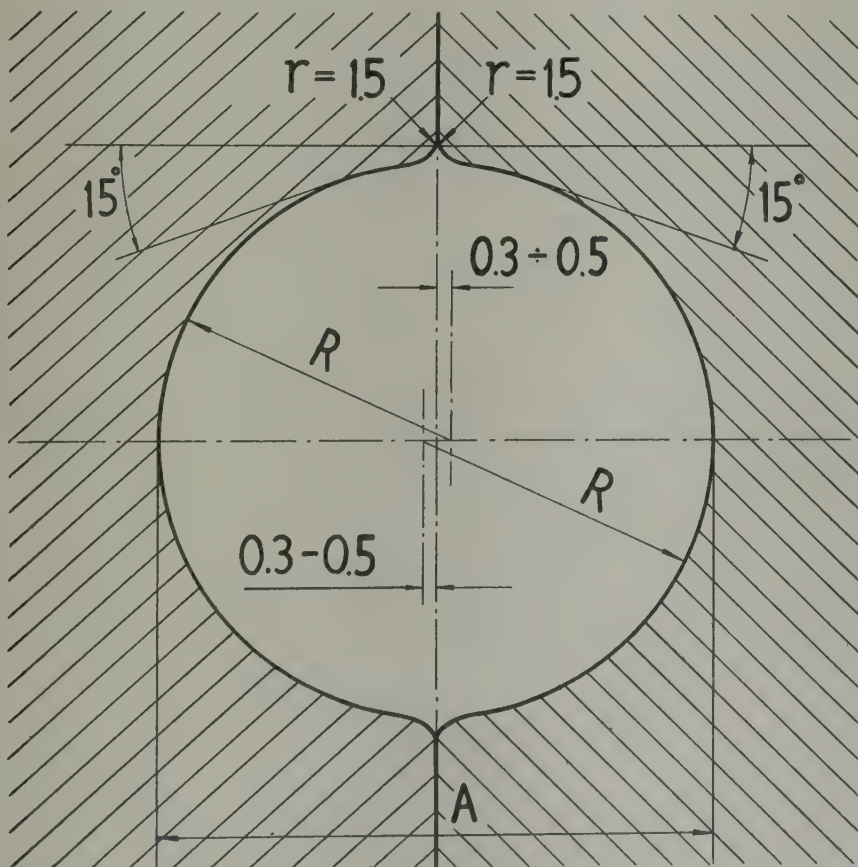


FIG. 29—CROSS SECTION OF GRIPPING IMPRESSION IN SET OF DIE FOR FORGING MACHINE: A is equal to the diameter of cold stock minus 0.3 to 0.5 mm.

Considerable importance should also be attached to the proper designs of the gripping impression of the die. When the die is closed the grip on the inserted bar should be sufficiently strong to avoid any sliding of the bar under the pressure of the punch. At the same time any damage of the stock by a gripping impression must be eliminated. The cross section of a well designed gripping impression is illustrated in Fig. 29.

## TOOLS FOR FLASH TRIMMING AND HOLE PIERCING

### Tools for Flash Trimming

The set of flash trimming tools consists of a trimmer blade and a trimmer punch.

A trimmer blade is designed and machined to the contour of the forging at the die parting line. For small and medium forgings trimmer blades are usually made from one piece of steel but for large forgings it is preferable to make trimmer blades built up from small segments mounted on the common base and accurately fitted together.

A trimmer punch which in most cases only pushes forgings through the trimmer blade is closely machined to fit the surface of the forging. The largest possible bearing surface is the most desirable as it eliminates the forging distortion which may appear in flash trimming operations. In contrast to the trimmer blade the trimmer punch generally does not require a sharp cutting edge.

Fixing of the trimmer blades used for small and medium sized forgings is usually made in special holders known as bolsters (Fig. 30) which are fastened by 4 bolts to the table of trimming press.

Fastening part of the trimmer punches should be designed according to the nature of the fastening elements provided on the slide of the press. In small trimming presses it is usually a round hole with a setting screw which takes the shank of the trimmer shown in Fig. 31. Bigger presses have a rectangular slot with several setting screws (Fig. 32). The large presses are usually made with a dove-tail slot and the fastening of the die is made on exactly the same principle as fixing the die block on the hammer.

The principle of design and standard dimensions of the tools for flash trimming are given in the drawings (Figs. 31 & 32). The angles of cutting edges of trimmer blades are different when provided for trimming the flash in hot or cold condition. When tools are designed for hot trimming they must be made with standard shrinkage allowance of  $1/100$ th of the nominal length.

### Tools for Piercing the Holes

Tools for piercing the holes consist of piercing blades and piercing punches (Fig. 33).

Piercing blades may be designed with usual cutting edges but quite often they are only serving as forging locating elements (Fig. 33B). In this case piercing operation is performed by a punch which gives rather a rough cut.

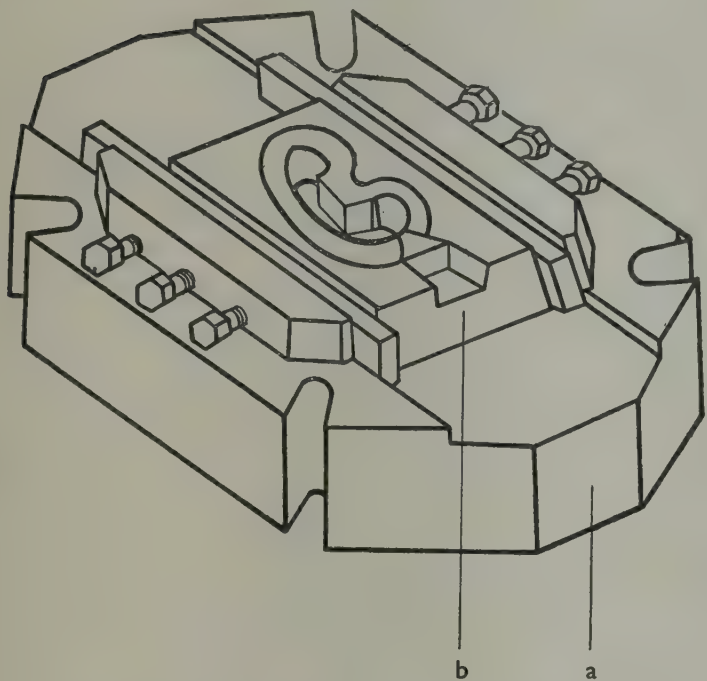


FIG. 30—TRIMMER TOOL SET WITH BOLSTER AND BLADE: (a) Bolster ; (b) Trimmer blade

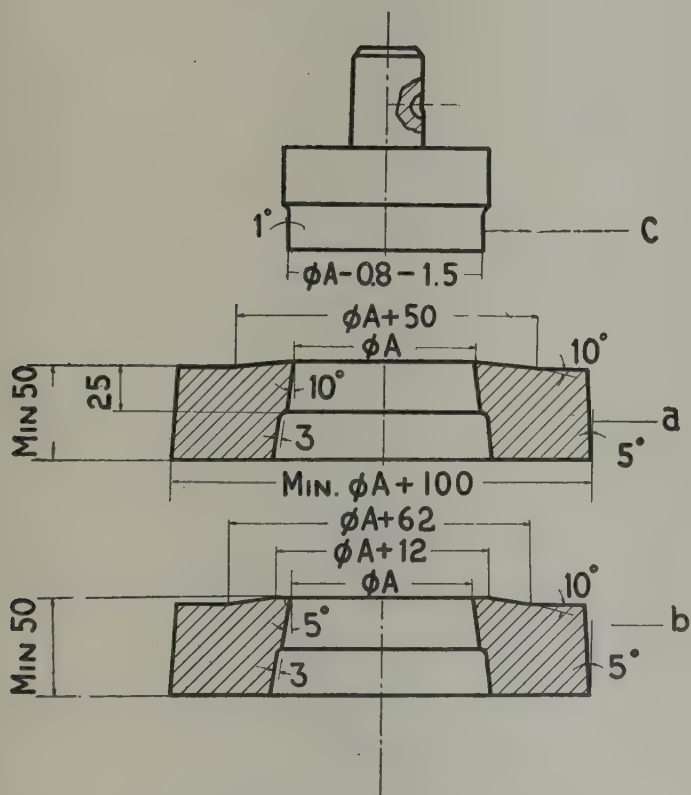


FIG. 31—DESIGN PRINCIPLES AND STANDARD DIMENSIONS OF TRIMMING TOOLS FOR ROUND FORGINGS: (a) Trimmer blade for hot work ; (b) Trimmer blade for cold work ; and (c) Punch common for hot and cold work

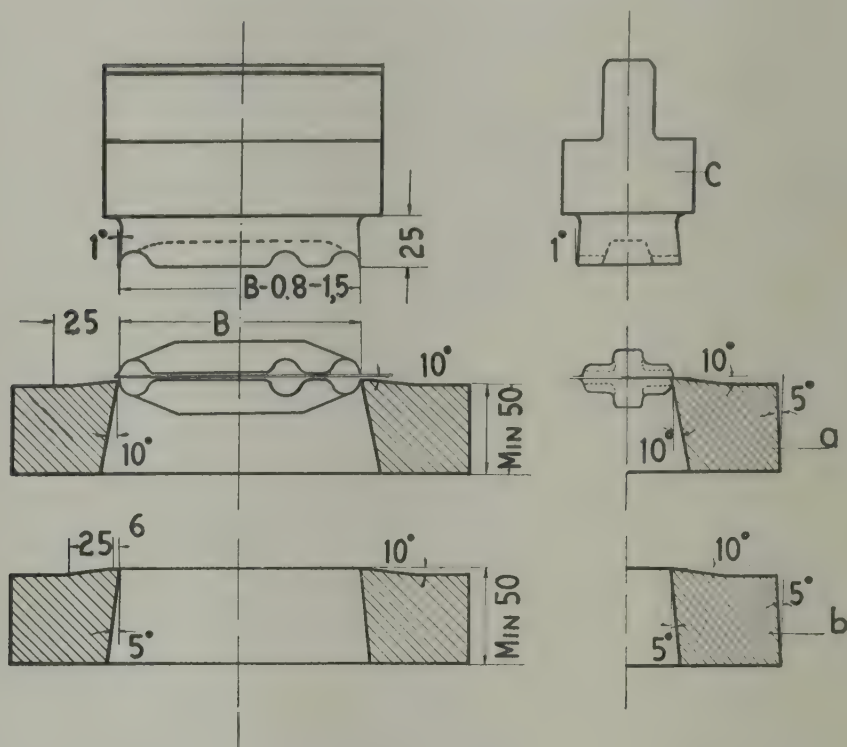


FIG. 32—DESIGN PRINCIPLES AND STANDARD DIMENSIONS OF TRIMMING TOOLS FOR LONG FORGINGS: (a) Trimmer blade for hot work; (b) Trimmer blade for cold work; and (c) Punch common for hot and cold work

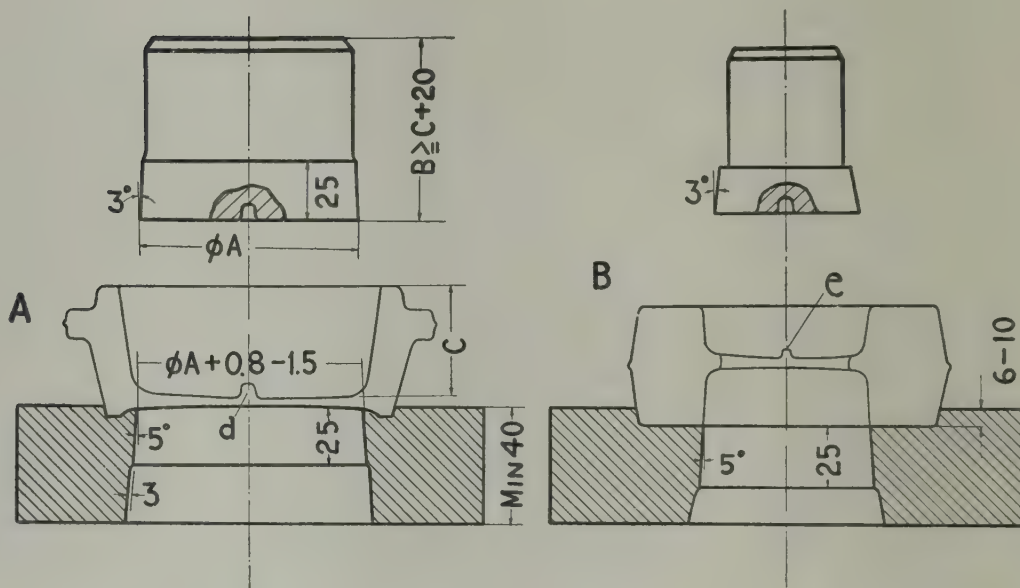


FIG. 33—DESIGN PRINCIPLES AND STANDARD DIMENSIONS OF THE PIERCING TOOLS: A—Trimming blade and loose punch; B—Locating stand and loose punch; and (d, e)=Pegs for locating of loose punch on the forging

Piercing punches must always possess sharp cutting edges. The piercing punches used in forge for cold work are very often simple loose punches (Fig. 33). This type of punch is put on each forging by hand. The punch

locates itself on small cylindrical pegs specially provided (Fig. 33 d, e). If the holes have an irregular shape two pegs are provided. The actual punching operation is done by a power press on which table a piercing blade is placed. The slide of the press in its down stroke movement presses the punch and pushes it right through. For hot piercing the punch is generally fastened to the slide of the press and the piercing blade is usually provided with a suitable stripper to strip forging from the punch.

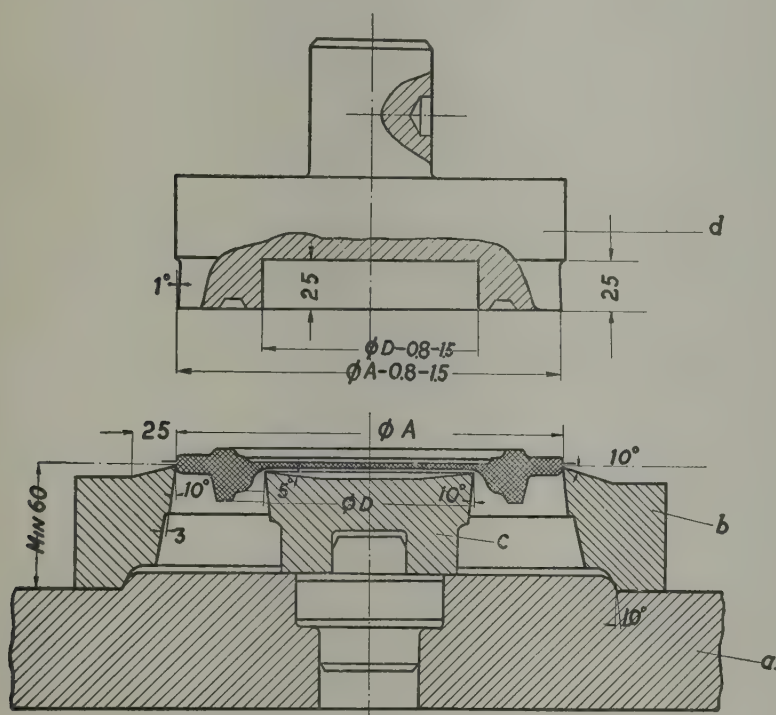


FIG. 34—COMBINED TRIMMING AND PIERCING TOOL  
DESIGNED FOR HOT WORK: (a) Base of the tool ;  
(b) Trimmer blade located by tapered recess ;  
(c) Piercing punch located on the peg ; and  
(d) Main punch

## Combined Trimming and Piercing Tools

Combined trimming and piercing tools which perform both operations simultaneously (Fig. 34) are designed with the trimming blade and punch located on a suitable base with tapered recess and central peg. When lifting up the punch and the trimming blade it is very easy to remove from the tool the finished, trimmed and punched forging.

# Electropolishing of Ferrous and other Alloys used in Internal Combustion and Mechanical Engineering

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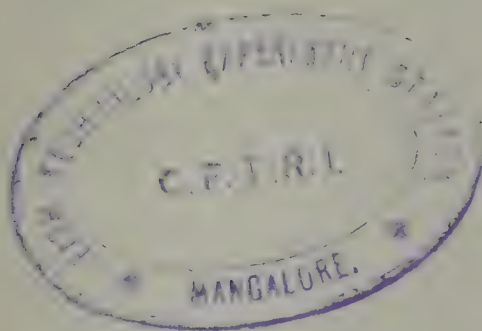
Electropolishing is the process by which a metal or an alloy surface is dissolved out anodically by carefully controlled conditions with regard to voltage, current density, temperature and electrolytes used. The process was first introduced by Jacquet of France as an auxiliary to metallography. The process is now commercially exploited for metal 'finishing' which gives to the surface a mirror-finish, with exceptional colour, tone and lustre. Other fields in which electropolishing has useful applications are in deburring, machining, increased adhesion of subsequent deposits, improved corrosion resistance, and polishing of inaccessible parts.

Electropolishing is being studied in many countries of the world, and in particular America, Germany, England, U.S.S.R., Japan and Australia. Materials polished on an industrial scale are among others, stainless steels, nichrome, chromel, copper, bronze and brass. A variety of products has received commercial attention: many automotive parts, household equipments, surgical and dental instruments, dies, mills, files, dairy and chemical process equipments, jewellery, novelties, etc.

World War II saw some striking advances in the electroplating of bearing surfaces. The silver-lead-iridium bearing, the best available for fatigue and thermal properties, was used extensively for highly loaded aircraft engine bearings. Silver, lead, and iridium were all deposited to required thickness—the iridium being later thermally diffused into lead. The study of the use of electropolishing in these steps will be of value. Another development in World War II has been the deposition of 'Porous Chromium'—thus facilitating the retention of lubricating oils in the bearing surfaces and increasing the life of these surfaces.

In certain metal finishing operations the metal cut (finished) will be influenced to appreciable depth below the surface. The resulting highly stressed, cold-worked surface layers, will be more susceptible to corrosive attack than the base metal. Minute surface cracks may also be formed which may greatly alter the fatigue life of the finished surface. Thus the exact machining process used to produce a surface may be of far greater importance with regard to subsurface metallographic transformations than with regard to surface finish produced. It is very significant to note that the electrolytically polished surfaces are particularly free from such subsurface changes.

The role of electropolishing in mechanical engineering deserves careful study. Published literature on electropolishing in relation to engineering is very meagre and very little work has been done in India on this subject.





## Fuel and Combustion

The second session was held on 6 April 1952 at 9 A.M. Dr. J. W. Whitaker, Director, Fuel Research Institute, Dhanbad, was in the chair.

Nine papers were read and discussion followed the reading of each paper.



# Some Problems in the Production of Power Alcohol

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Among the annually renewable and practically inexhaustible liquid fluids which are particularly adaptable for internal combustion engines, power alcohol represents one of the richest and the most competitive sources. The raw material for its production is widely and uniformly distributed throughout the world unlike petroleum whose occurrence is confined to the more fortunate parts of the globe.

So long as the sun continues to shine and promote the photosynthetic activity of the chlorophyllous plants, there will be no dearth of the material in one form or another. Most of the raw materials are drawn as by-products of the sugar, lumber and agricultural industries.

One of the inexpensive, readily available, and easily convertible raw materials for alcohol production is the molasses which constitutes the inevitable by-product of the sugar industry. Other raw materials, which are available in larger quantities, are the cellulosic wastes from the lumber and the farm industries ; but these raw materials contain the alcohol yielding sugar in a difficultly available form and demand, for its release, expensive and difficult processings involving the employment of expensive capital equipment.

Practically the entire output of power alcohol which is produced in India today is from molasses. There are no synthetic alcohol plants in this country.

The country's output of alcohol in its 15 factories for 1950 amounted to about 4.5 million gallons as against 4.2 in 1949 and 3.8 in 1948. The factories have not been able to work to their full capacity, and the planned targets have never been attained. The efficiency of the plants are low and the costs of production are not competitive. An account of the causes which have led to this inefficiency has been given and remedial measures for improving their efficiency and for cutting down the costs of production to competitive levels, discussed. The threatened competition from synthetic alcohols has been indicated.

# Utilisation of Heavy fuels in High Speed Diesel Engines

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## Introduction

Diesel engines may be classified into three groups:

1. Large low speed engines with 10 in. bore and over and a speed range of 150 to 350 r.p.m.
2. Medium size, medium speed engines, with 6-10 in. bore and a speed range of 400 to 700 r.p.m.
3. Small high speed engines with 6 in. bore and a speed of 1,000 r.p.m. and above.

Class (1) engines usually run on fuels with viscosities ranging from 100-500 sec. (Redwood No. 1). But such engines have been run on still heavier fuels. One notable example is that of the Anglo-Saxon Petroleum Co., Ltd., running some of the large marine engines used in tankers on heavy fuel with a viscosity of 3,000 sec. (Redwood No. 1).<sup>1</sup>

Class (2) engines normally run on light diesel oil (L.D.) which has a viscosity range of 40-100 sec. (Redwood No. 1). Even in this case heavier fuels with a viscosity range of 350-750 sec. have been tried at Thornton Research Centre and elsewhere.<sup>2</sup>

Class (3) engines normally run on gas oil or high-speed diesel oil (H.S.D.) with viscosity less than 45 sec. (Redwood No. 1). Experiments are taking place to replace this fuel by light diesel oil.

The Department of Internal Combustion Engineering at the Indian Institute of Science has interested itself, partly at the instance of Messrs Kirloskar Oil Engine Co., in the utilisation of light diesel oil and somewhat heavier fuels in high speed diesel engines.

The work has been divided into (a) study of combustion and performance of a high speed diesel engine running on different fuels and fuel blends and (b) a study of the wear properties of engines running on different fuels. This paper gives briefly the work done under (a) and the results obtained.

A Kirloskar Petter Engine type AV1 and a Ricardo E-6 engine were used for the tests. The fuels used were H.S.D. oil, L.D. oil and a blend of L.D. oil with 5 per cent alcohol. Alcohol blending was included for two reasons: (1) to find out the influence of alcohol on the combustion of hydrocarbon fuels and (2) to reduce the viscosity of the heavier fuel so as to make it flow more easily.

The properties of the H.S.D. oil and L.D. oil used in this investigation and also B.S. specifications for similar fuels, are given in Table 1.

TABLE I—PROPERTIES OF H.S.D. AND L.D. OILS

	Available and used in the investigation		British standard specifications	
	H.S.D. oil	L.D. oil	B.S. 209 1947 Class A	B.S. 209 1947 Class B
Sp. gr. at 15°C. ...	0.84	0.88	—	—
Flash point °F. ...	190	150	150	150
Viscosity at 100°F. sec. } (Redwood No. 1) ...	35	45	31/45	100 max.
Carbon Residue Conrad- son (% wt.) ...	0.05	1.0	0.1	2.0
Sulphur (% wt.) ...	0.7	1.5	1.5	2.0
Water (% vol.) ...	Nil	0.05	0.1 max.	0.25 max.
Sediment (% wt.) ...	Nil	0.01	0.01	0.1
Ash (% wt.) ...	Nil	0.02	0.01	0.03
Pour Point °F. ...	10	50	—	—
Cal. value B.t.u./lb ...	19,500	19,100	19,000 min.	18,500 min.

The details of the Kirloskar Petter Engine are given below:

No. of cylinders ...	...	...	...	1
Bore ...	...	...	...	3.15 in.
Stroke ...	...	...	...	4.33 in.
r.p.m. ...	...	...	...	1,500
Rated h.p. ...	...	...	...	5
Compression ratio ...	...	...	...	19:1
Lubricating oil used ...	...	...	...	SAE-30
Injection pressure ...	...	...	...	2,500 lb./sq. in.
Injection nozzle ...	...	...	...	Single hole 0.46 mm. diam. 0.90 mm. long
Injection commences ...	...	...	...	27° before T.D.C.
Pre-combustion chamber type (Fig. 1)				

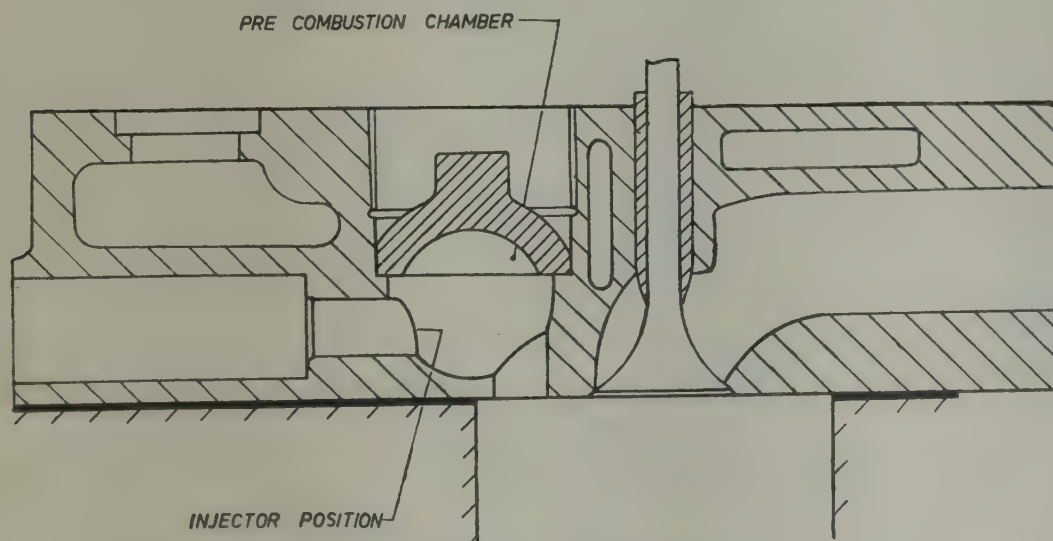


FIG. 1—COMBUSTION CHAMBER OF KIRLOSKAR PETTER OIL ENGINE

### Test Procedure

During every test the engine was warmed up for one hr., the water temperature was maintained at  $80^{\circ}\text{C}$ . and the oil temperature between  $138^{\circ}\text{F}$ . and  $145^{\circ}\text{F}$ . depending on the load. Fuel consumption was noted periodically and the performance of the engine at part and full load conditions and 8-12 hrs. endurance runs was studied. This procedure was adopted for all the three fuels, i.e., H.S.D. oil, L.D. oil and alcohol blends. The engine was run for about 25 hrs. in all on each fuel.

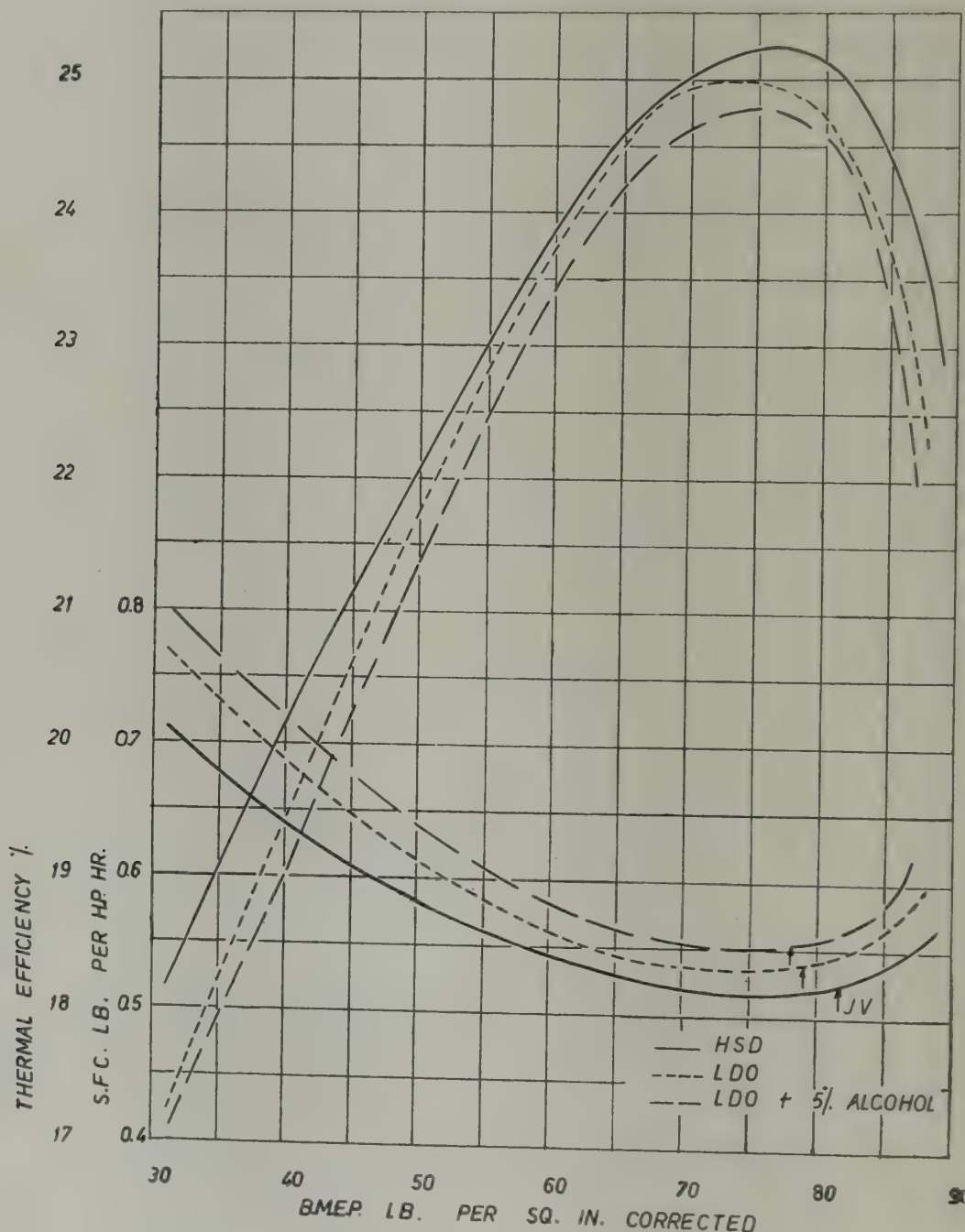


FIG. 2—PERFORMANCE OF DIESEL ENGINE RUNNING ON DIFFERENT FUELS AND FUEL BLENDS

## Test Results

The various performance curves are shown in Fig. 2 which indicate that

- (1) There is little difference in the maximum output of the engine while using the three fuels.
- (2) Thermal efficiency is only slightly less for the L.D. oil and the blend.
- (3) B.M.E.P. at just visible exhaust is only 3 per cent less for the L.D. oil and the blend.

No excessive carbon deposits on the nozzles were found in any of the three cases, nor was any hard carbon found on that portion of the walls of the precombustion chamber where the fuel might be expected to impinge. Slight carbon deposits were found in all the cases only on that part of the piston top away from the throat of the precombustion chamber. This may perhaps be due to less air movement in that region. It is also to be noted that in spite of the higher viscosity and the higher carbon residue of the light diesel oil, no hard deposits were found on the inside of the precombustion chamber. It appears that more Conradson carbon residue than is at present allowed for by B.S. 209—1947 for class 'A' diesel oil for high speed diesel engines is permissible, especially if there is considerable air movement in all parts of the combustion chamber, particularly where there is a likelihood of fuel cracking. Injection pressure in this engine is rather on the higher side of the average value for a similar engine and this may have contributed towards the efficient combustion of light diesel oil.

A serious difficulty with alcohol was that it separated from the fuel unless the blend was kept stirred periodically. Higher percentages of alcohol could not be tried on account of this.

## Test with Ricardo E-6 Engine

The effect of varying the injection timing on the combustion of the three types of fuels was next tested in a Ricardo E-6 Engine.

The details of the Ricardo E-6 Engine are given below:

Bore	...	...	...	...	3 in.
Stroke	...	...	...	...	4.375 in.
r.p.m.	...	...	...	...	1,000/3,000
Compression ratio	...	...	...	...	20:1
Injection pressure	...	...	...	...	1,500 lb./sq.in.
Precombustion chamber of the					
Ricardo Comet Mark II type					
Lubricating oil used	S.A.E.-30				

## Test Procedure

In this case, the tests were conducted with three different injection timings for each fuel, namely, the start of static injection was (i) 21° before

T.D.C. (ii)  $37^\circ$  before T.D.C. (iii)  $45^\circ$  before T.D.C. The results obtained are shown in Figs. 3, 4 and 5. For an injection at  $37^\circ$  and  $45^\circ$  the performance of the engine on H.S.D. or L.D. oil are so nearly the same that the curves follow one another very closely. The optimum performance was in case (ii). However, when the injection start is at  $21^\circ$ , combustion with L.D. oil deteriorates resulting in higher fuel consumption, smoky exhaust and low power output and higher exhaust temperatures. All this may be due to the lesser reactivity of L.D. oil.

A blend of 5 per cent alcohol and 95 per cent of L.D. oil had performance features similar to straight L.D. oil. No special trouble was encountered in starting nor was there any missing even at light loads with start of injection  $21^\circ$  before T.D.C.

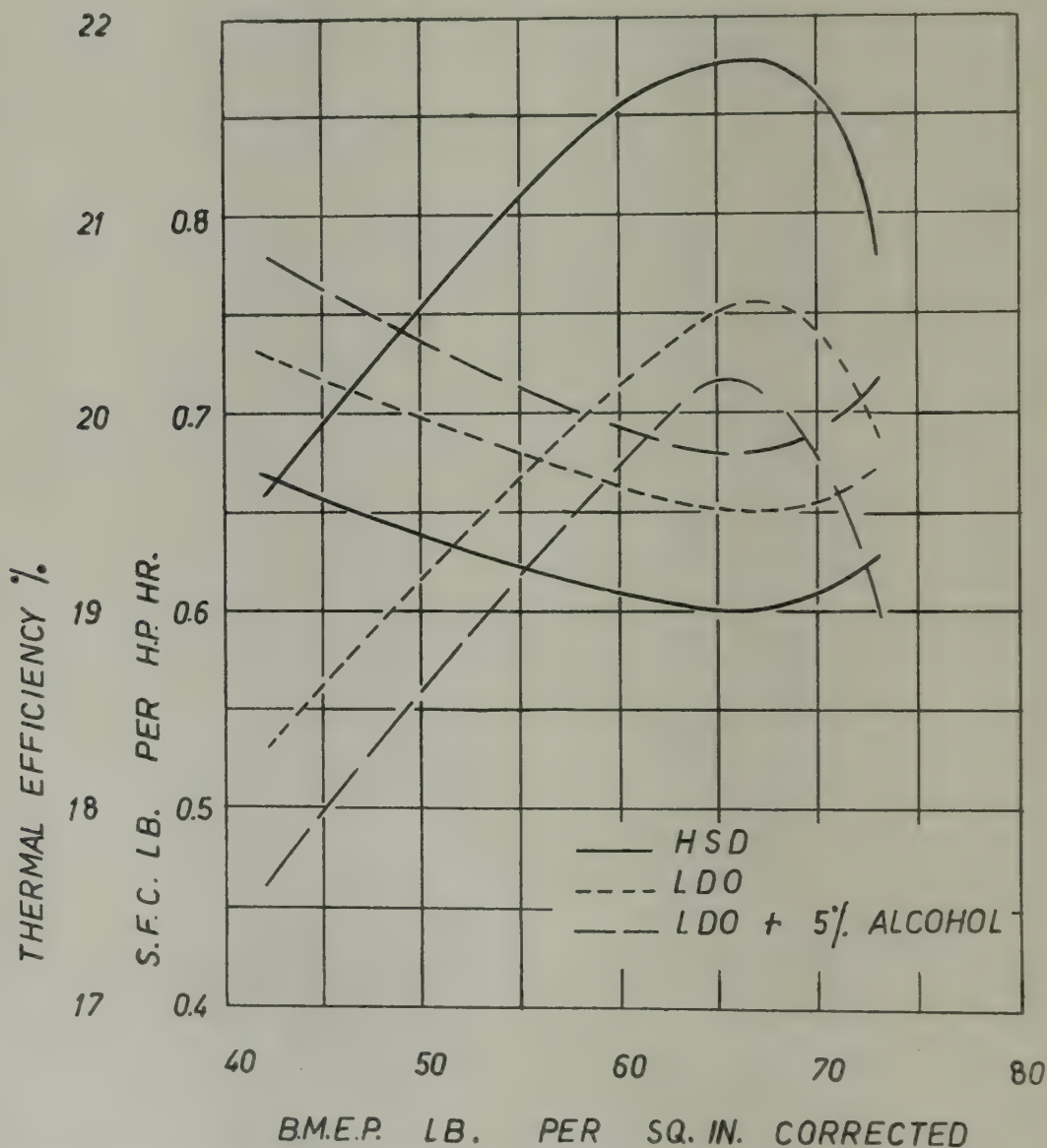
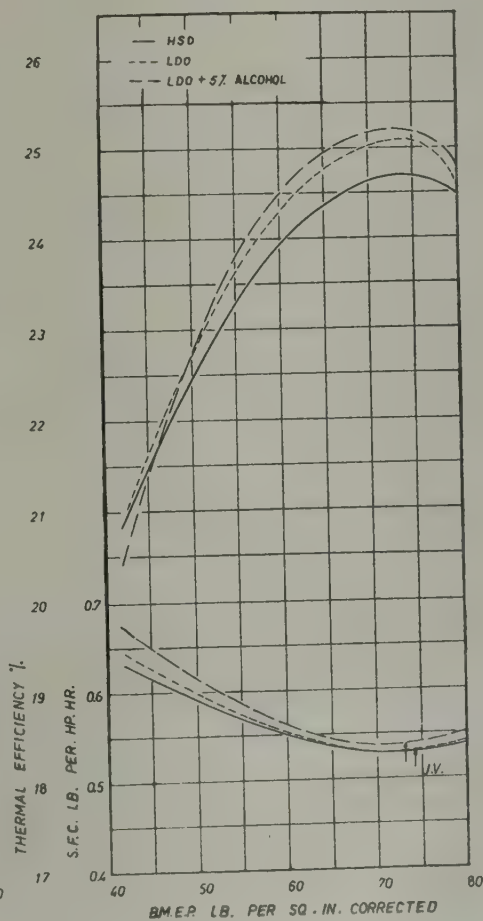
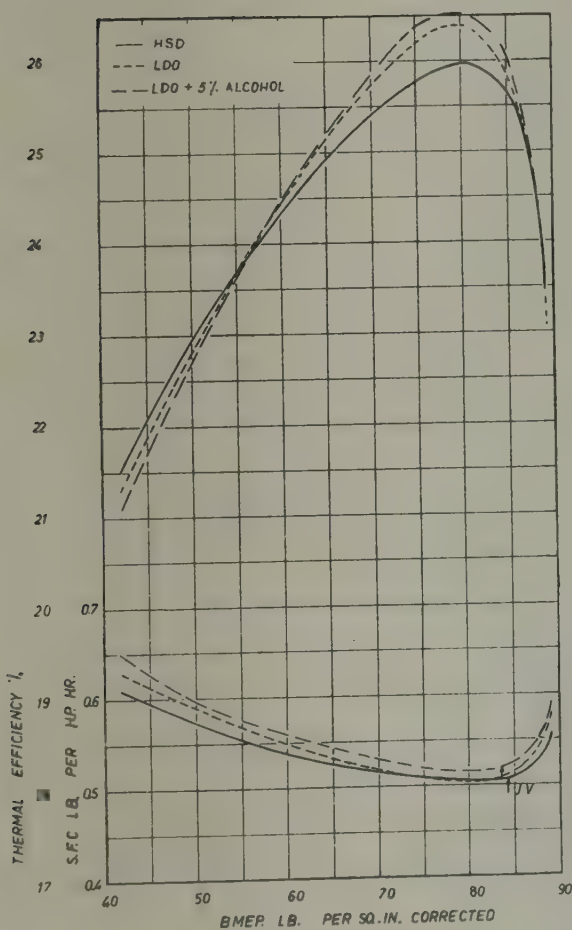


FIG. 3—PERFORMANCE OF DIESEL ENGINE RUNNING ON DIFFERENT FUELS AND FUEL BLENDS

## Conclusions

1. Experiments conducted here and elsewhere have indicated that heavier fuels can be burnt successfully in diesel engines.
2. During short trials conducted, there have been no excessive or harmful carbon deposits.
3. Maximum B.M.E.P. for clear exhaust is only slightly lower for heavier fuels.
4. Blending of either H.S.D. oil or L.D. oil with suitable percentages of alcohol, has not shown any appreciable difference in the performance of the engine.
5. Alcohol may be more beneficial when blended with still heavier fuels or vegetable fuels with high viscosity. Work in this direction may be extended.
6. Endurance and wear tests are the next logical steps in finding out more about these fuels and work in that direction may prove fruitful.



FIGS. 4 (left)—& 5 (right)—PERFORMANCE OF DIESEL ENGINE RUNNING ON DIFFERENT FUELS AND FUEL BLENDS

Finally there seems to be a general tendency towards the utilisation of heavier fuels and higher injection pressures in high speed diesel engines. Much trouble may not be encountered in the injection of class 'B' fuels. But an increase in viscosity is inevitably accompanied by increases in the values of Conradson carbon residue, sulphur content and other impurities. It is extremely difficult, if not impossible, to ascribe the inefficient performance of a fuel to any one of these causes. However, as mentioned earlier, a higher Conradson residue than is at present allowed for high speed diesel oils seems to be permissible. Whether sulphur content has considerable effect on the wear of an engine is a much debated point. Chromium plated top piston rings are said to have decreased the wear of not only the cylinder but also the lower piston rings. Experiments are in progress in this laboratory on various aspects of wear of engines and some valuable data may soon be available.

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# Investigation on the Possibilities of Gas Generation in Cyclonic Chambers

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## I. Introduction

One of the main factors that require attention in the case of thermal power plants used both for stationary and transport purposes in recent days is the problem of devising suitable combustion equipment for burning low grade liquid and solid fuels. The resources of low grade coals in India are vast when compared with good quality coals<sup>1</sup> and it is but proper to investigate the various possibilities of utilising these fuels for power generation so that the good quality coals can be conserved for metallurgical operations. The two main difficulties encountered with these fuels are (a) low rates of combustion and (b) high ash content. A high rate of heat release can be obtained with solid fuels if they are burnt in the pulverised form, since a far greater surface area is available for mixing up with the oxygen, but the combustion apparatus should be so designed that the individual fuel particle is kept in the chamber till it is reduced to a very small size, thus ensuring a high degree of completion of combustion. Recent developments have indicated that vortex type of combustors satisfy the above conditions.<sup>2</sup>

However, when such apparatus is used in conjunction with gas turbines, the residues from the combustion products may have an adverse effect on the blades, unless the combustion gases are cleaned before their entry to the turbine section. One way of obviating this difficulty is gasification of the fuel as this method lends itself to the process of cleaning.

Considering the process of gasification, the difficulty with the present day gas plants for high power generation is their bulkiness due to their low rates of gasification—a result of the fuel being in lump form. The technique of gasification has indicated that the rate of gasification can be vastly improved if the fuel particle size is reduced to as small a figure as practicable and if the velocity of gas relative to the particles is increased to a high value.<sup>3</sup> The maximum relative velocity of a particle in suspension is equal to its terminal velocity, if we disregard the influence of additional forces in non-homogeneous velocity fields. A superfine char particle ( $1\ \mu$ ) has practically no velocity relative to its surroundings and consequently follows any movement of its carrier gas. As mass transfer is confined within the possibilities of diffusion, the enormous increase in surface is not effective. However, the terminal or settling velocity can be increased theoretically to any extent, if the particle

moves steadily in a circularly rotating gas stream. This is achieved in a cyclone chamber and thus the apparent difficulty of combination of smaller particle size with high relative velocity can be overcome.

## 2. Working Principle of Cyclone Chamber

The principle of the cyclone chamber was introduced by Lander<sup>4</sup> and has been developed by different research workers.<sup>5,6</sup> As the cyclone gas producer deviates from the cyclone combustion chamber only in certain final conditions, the theory will be developed for the combustion chamber and the modifications required for adoption to a gas producer will be indicated at a later stage. In principle, the design is based on the following:

(a) Combustion in the vortex—The normal form of cyclone combustion chamber is shown in Fig. 1. Air and fuel are admitted tangentially and enter a vortex type of flow through its periphery. The combustion gases are extracted axially from the centre. Combustion takes place between periphery and centre of the vortex. It is started in the primary combustion zone, situated in the tangential entry of the chamber, where the primary flame is stabilised. Part of the fuel particles gets ignited in the primary chamber and then with the remaining particles, they complete their combustion in the vortex of the cyclone chamber.

(b) Equilibrium conditions and dynamics of motion for fuel particles—At a particular radius within the vortex, the fuel particles are held under the influence of centrifugal forces acting towards the periphery and the viscous drag directed towards the centre. At equilibrium the two forces will be equal and then the droplets will rotate at a constant radius, called the equilibrium radius about the centre under the action of these two forces with relative motion of the gas if their size would be constant. Thus fresh oxygen is continuously supplied to the periphery of the fuel particle by the relative motion of the gas with respect to the particle. An interesting feature of the chamber is the absence of random turbulence which is usually necessary in normal combustion chambers for raising the rate of combustion, and therefore the pressure losses in cyclone chamber should be small.

Mathematical investigations have indicated that the equilibrium of the droplet can be made stable by suitably shaping the walls of the cyclone chamber.<sup>7</sup> Thus any deviation from the equilibrium radius gives rise to forces which lead the droplet back to its equilibrium radius. Within such a chamber droplets of different sizes find equilibrium conditions at different radii; big droplets find equilibrium conditions at the periphery and small droplets near the centre of the chamber. Thus the droplet approaches the centre in spiralic motion in the course of its combustion. Under the restraint of the forces acting on it, the droplet or particle will also oscillate radially about its equilibrium radius and this oscillation results in a rhythmic increase of the relative velocity of the gas with respect to the droplet.

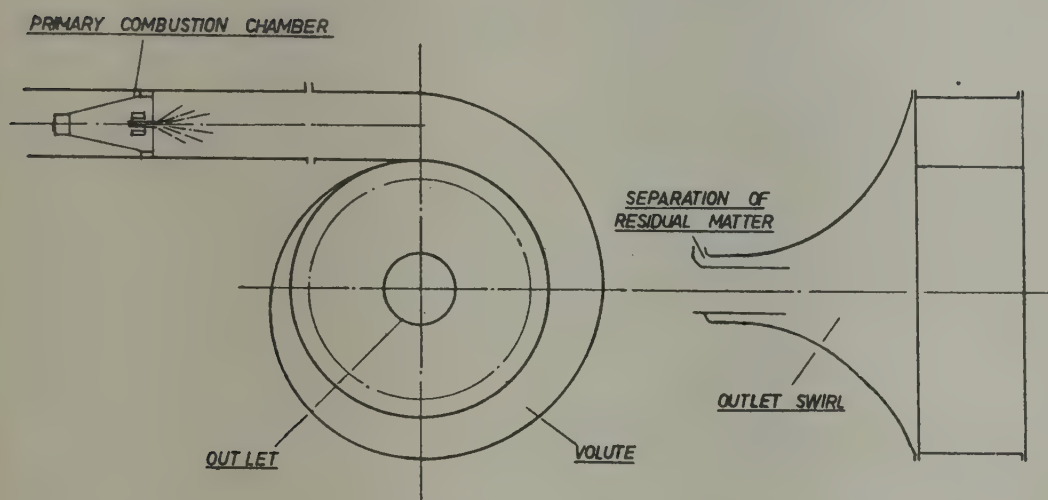


Fig. 1—DIAGRAM OF CYCLONE CHAMBER

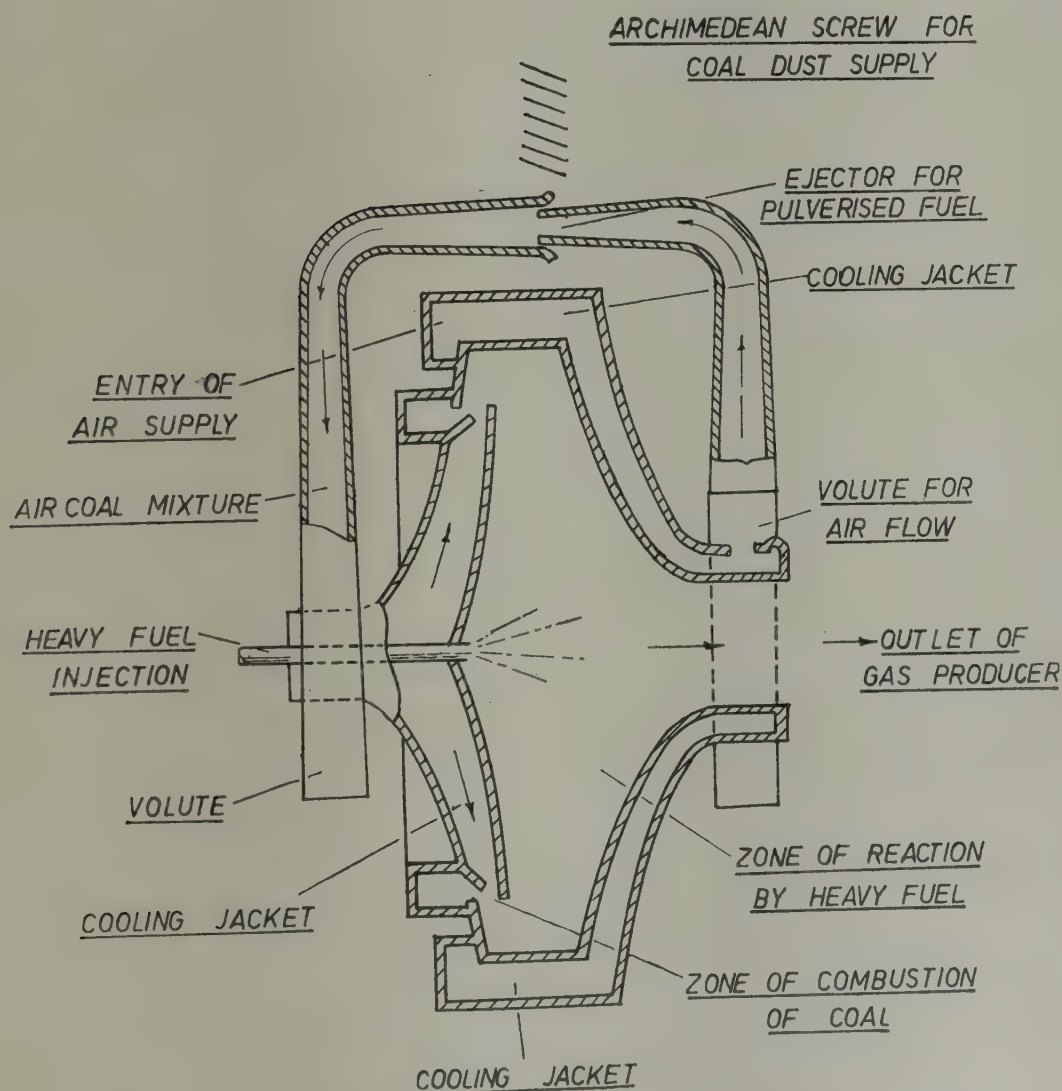


FIG. 2—CYCLONE GENERATOR FOR PULVERISED FUEL AND LIQUID FUEL

### 3. Fundamental Requirements for Producer Gas Formation

Since the changes effected in a cyclone chamber for gas production are based on the theory of gasification, a brief analysis of the chemical process of gasification will help to understand the changes.

Gasification is a reaction between solid carbon and an oxygen carrying gasifying agent, the blast, e.g., air, steam, carbon dioxide or mixtures thereof, yielding combustible gases ( $\text{CO}$ ,  $\text{H}_2$ ,  $\text{CH}_4$ ). While combustion takes place with excess oxygen, or at least with the theoretical oxygen requirement (stoichiometric or air-saturated combustion), gasification takes place with excess carbon. However, the gasification process should by no means be considered as some kind of incomplete combustion, rather, it includes oxidation as well as reduction processes. The temperature of process should be more than  $800^\circ\text{C}$ . to obtain a high percentage of  $\text{CO}$ .

The following conditions should be satisfied for efficient gasification:  
 (a) two distinct zones of reaction, namely, oxidation and then reduction and  
 (b) the resultant gases of oxidation process should be at a temperature higher than  $800^\circ\text{C}$ .

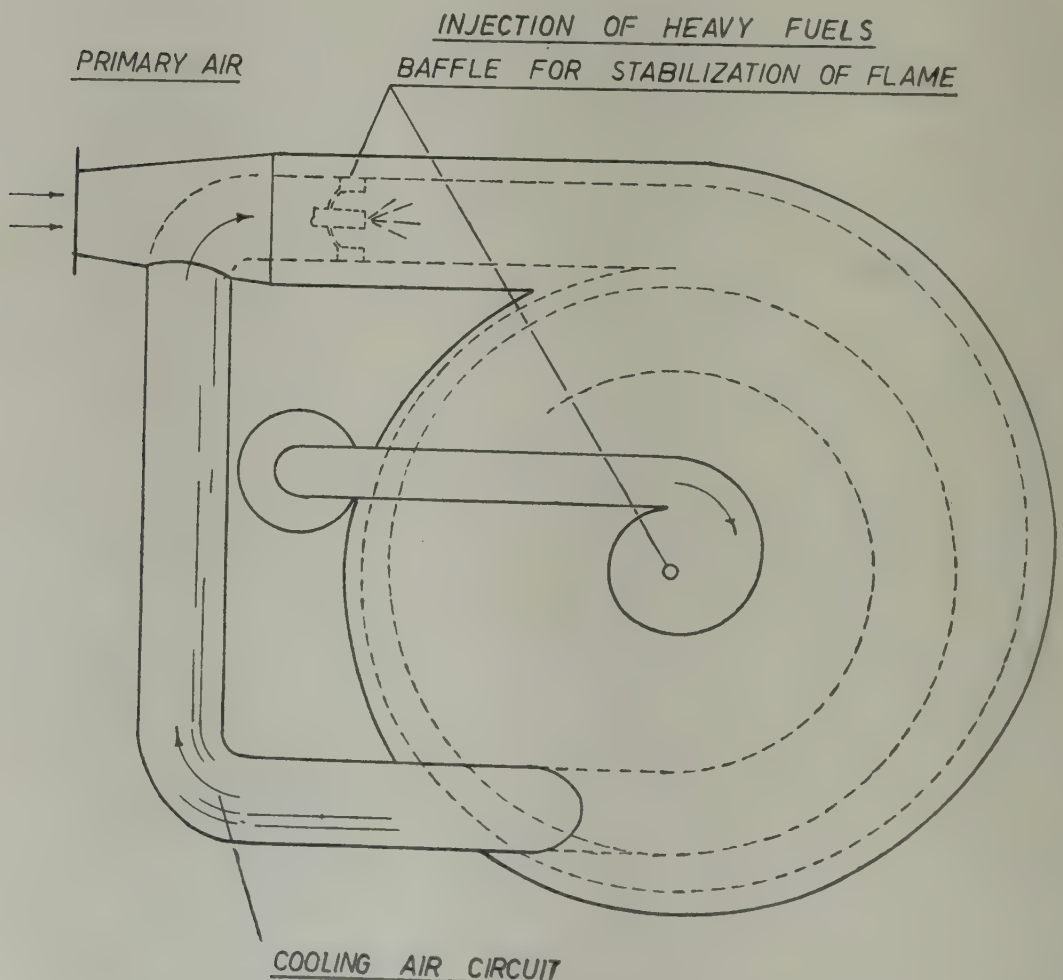


FIG. 3—CYCLONE GENERATOR FOR LIQUID FUEL, DUAL SUPPLY

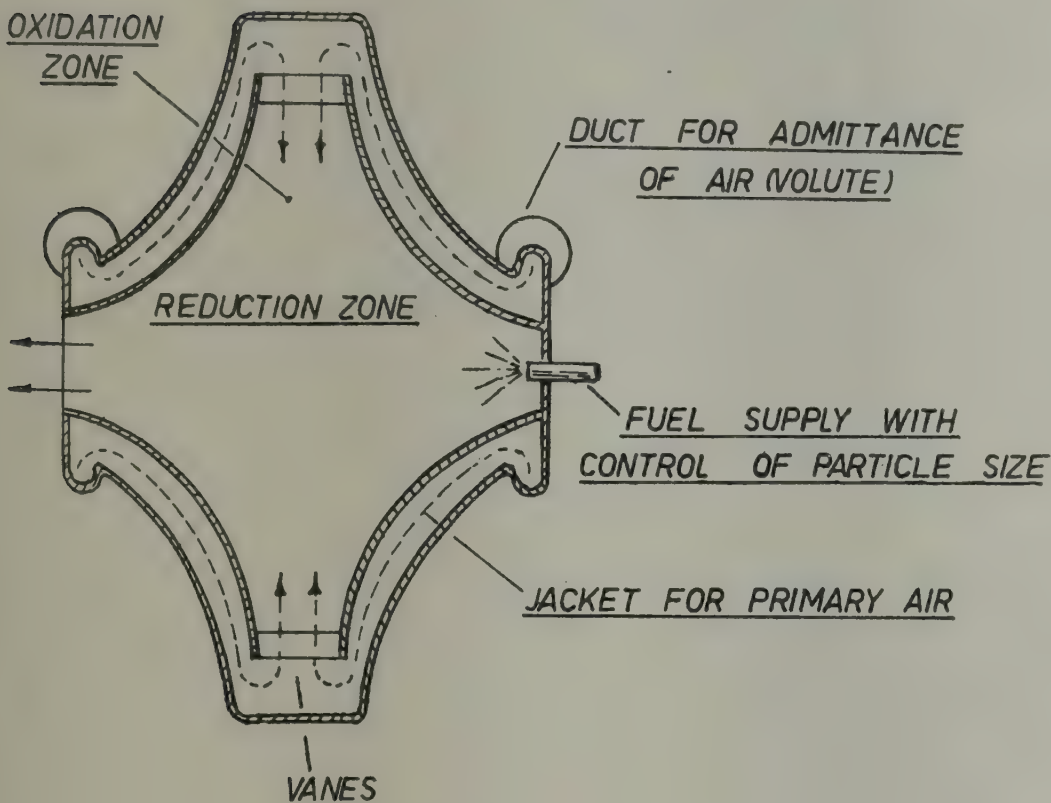


FIG. 4—CYCLONE GENERATOR FOR SINGLE FUEL SUPPLY

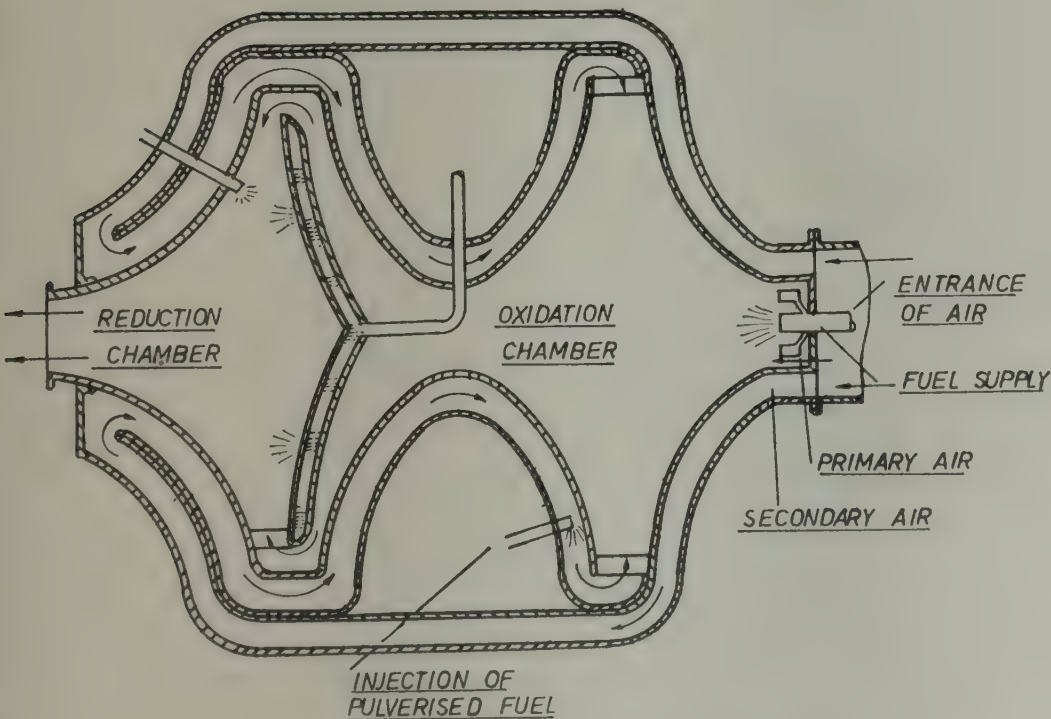


FIG. 5—TWO CHAMBER CYCLONE GAS PRODUCER

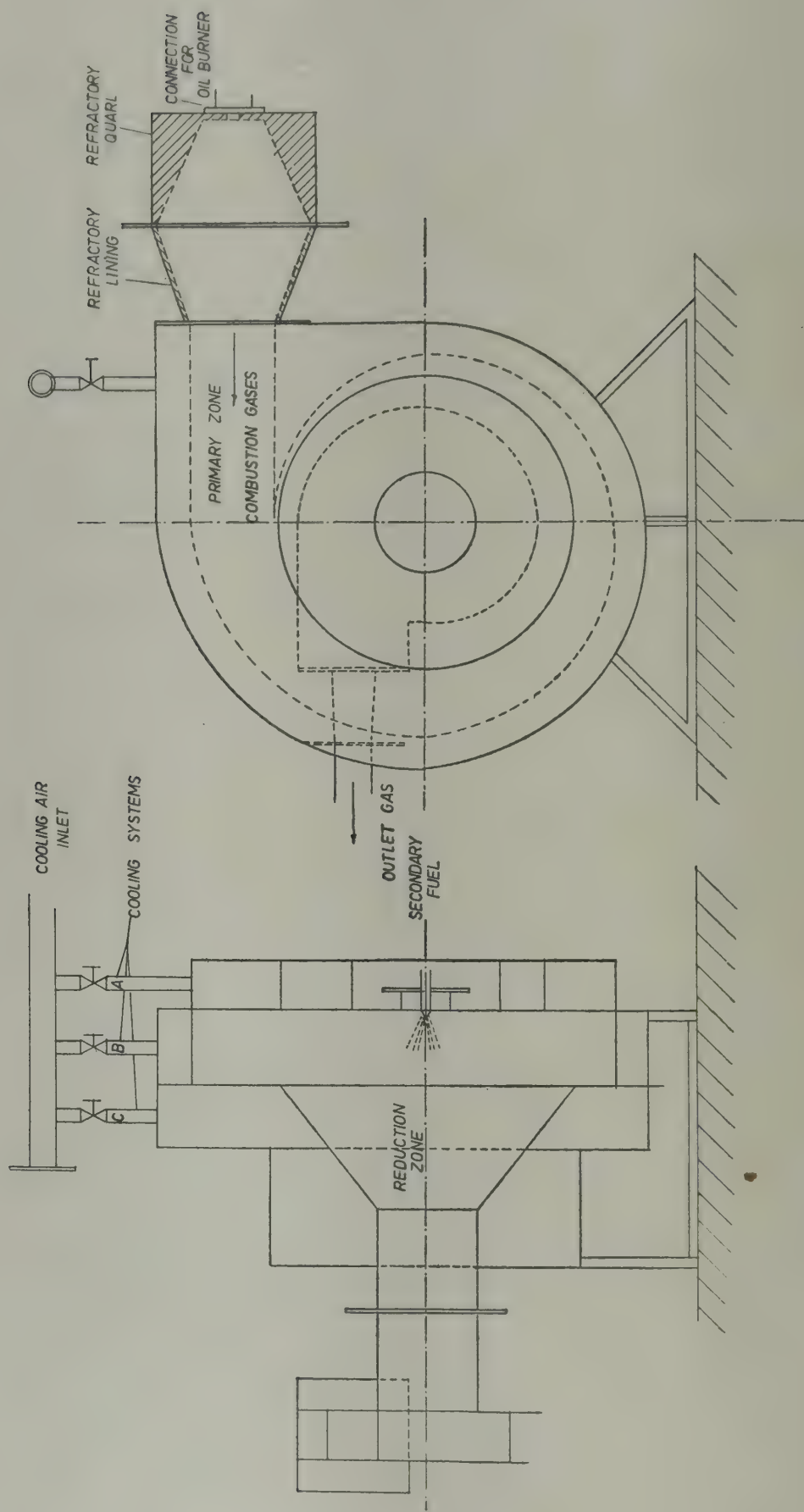


FIG. 6—DESIGN DETAIL OF CYCLONE GAS GENERATOR

#### 4. Application of Cyclone Chamber for Gas Generation

There exist two main possibilities of design to effect gasification in cyclone chambers:

- (1) Oxidation and reduction carried out in one cyclone chamber only ;  
and
- (2) Oxidation and reduction carried out separately in two different cyclone chambers.

Figs. 2 to 5 represent the various methods of application based on the above mentioned processes.<sup>8</sup>

#### 5. Programme of Experimental Work undertaken in I.C.E. Department

A cyclone gas producer based on the design principle indicated in Fig. 3 has been built for a mass flow of  $\frac{1}{4}$  lb./sec. Fig. 6 represents the design details of the experimental set up. To obtain the necessary high temperature for the reaction zone, kerosene will be utilised for the primary zone of the fuel system ; furnace oil will be adopted for the secondary zone. A gear pump has been used as a fuel pump for the secondary zone and preheating will be resorted to, to reduce the viscosity of the oil to the desired level. Unlike in the case of the combustion chamber, the temperature decreases with decreasing radius and hence a certain instability of vortex might result. Gas analyses are expected to indicate the probable reactions taking place inside the chamber.

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# Utilisation of Indigenous Fuel in Combustion Power Plants

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Different procedures must be followed in dealing with the different types of low grade and waste fuels of India to obtain the maximum fuel value from them economically. Lumps of low grade coal or low grade coal fines briquetted with pitch, or lignite briquetted with pitch can be subjected to low temperature carbonisation and the residual cake gasified to obtain not only synthesis gas for the production of liquid fuels but also hydrogen for the hydrogenation of tar to produce liquid fuel. Low temperature coke can also be employed as a substitute for petrol in buses and lorries. Saw dust, wood chips, etc., can be gasified to produce synthesis gas or hydrogen for use in the production of liquid fuels. Molasses and *mohwa* flowers or agricultural wastes like paddy husk, bagasse, cereal straws, etc., after hydrolysis with dilute acids, can be fermented to produce alcohol and power alcohol can be employed as a substitute for petrol or used in admixture with petrol. Natural gas and methane recovered or synthesised from by-product gases or produced by bacterial decomposition of waste cellulosic materials can be employed either in the compressed state or as liquid as an efficient substitute for petrol in internal combustion engines.

# Utilisation of Vegetable Oils as Fuels for Compression Ignition Engines

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## Introduction

India being the second largest producer of vegetable oils in the world coming only next to the U.S.A., vegetable oils naturally loomed as a possibility for compression ignition engines. Groundnut (peanut) oil of which India is the world's largest producer, coconut oil, gingelly oil and castor oil were some of the vegetable oils that were studied for use as engine fuels.<sup>1,2</sup>

Utilisation of vegetable oils as fuel for compression ignition engines has been advocated many decades back by Dr. Diesel.<sup>3</sup> He advocated the use of arachis or groundnut oil as fuel for the diesel engine.

Constam and Schlapfer<sup>3</sup> conducted tests on arachis and palm oils with a view to their use in diesel engines and found them satisfactory. H. Moore<sup>3</sup> tried whale oil and found no difficulty in obtaining excellent running with no modification of engine settings. Baker and Sweigert<sup>4</sup> have investigated the use of groundnut oil, soya bean oil and cotton seed oil as fuel for compression ignition engines.

Baker and Sweigert of U.S.A.<sup>2</sup> used a Fairbanks Morse single cylinder diesel engine of  $4\frac{1}{2}$  in.  $\times$  6 in. and a compression ratio of 14.5:1.0—the rated horse power being 10 h.p. at 1,200 r.p.m. This engine was run on diesel oil, peanut or groundnut oil, soya bean oil and cotton seed oil. They observed that the load range for vegetable oils was lower than the load range for diesel oil and also that engine performance with groundnut oil was better than with either soya bean or cotton seed oil. The fuel consumption per h.p. hr. with groundnut oil was less than with either of the other two vegetable oils. They opine that vegetable oils having lower heating value and different combustion characteristics tend to make the load range lower. If the engine is re-designed to meet the characteristics of the vegetable oils, the load range can be extended.

Bender<sup>3</sup> has presented some results on the use of vegetable oils as fuel for diesel engines in French West Africa. A Berlick truck equipped with a Ricardo Diesel engine having a comet turbulence chamber was run on peanut oil as fuel. The injection pressure was raised from 1,422 psi to 2,350 psi to get better atomization of the peanut oil. The truck completed 10,000 miles of running. The engine was taken apart every 5,000 miles for check up and the following observations were made:

1. Combustion chambers and piston heads were covered by a very light, non-adhesive soot similar to what is found with mineral oil.

2. Excellent condition of injector nozzles, no deposits on needles or seats.
3. Excessive deposits of carbon in the ring grooves of the upper and second rings ; none in the third. None of the rings got stuck.

To avoid ring sticking when using vegetable fuel, Bender suggests that the engine should be started on mineral oil and operated until the engine had reached a temperature high enough to withstand gum formation in the ring grooves. To avoid carbon formation at the nozzle, the fuel was preheated to a temperature of 175°F. by means of the exhaust gases so that its viscosity become comparable to that of a mineral fuel.

Tests in India and abroad clearly demonstrated the possibility of using groundnut oil as fuel for diesel engines. However, the duration of the tests was considered insufficient to ascertain the deleterious effects if any on the engine. The Lister high speed diesel engine and the Cooper low speed diesel engine were run each for a thousand hrs. The results on the former<sup>3</sup> show that groundnut oil has no deleterious effects such as corrosion, pitting, etc. With groundnut oil as fuel, the engine must be decarbonised every 500 hrs. instead of 1,000 hrs. as recommended by the makers, and the valves should be cleaned every 200 hrs. to prevent sticking. Wear and tear were not excessive.

The results of the 1,000 hrs. runs may be summarised as follows :

### **Fuel Filters**

There was no trouble with the fuel filter at any time during the entire run and the flow of fuel was uninterrupted. However, the filter element was cleaned every 100 hrs. to prevent it being clogged by any matter present in groundnut oil.

### **Atomiser**

There was a tendency for carbon deposit to be formed at the atomiser tip which affected the startability. This necessitated the cleaning of the atomiser on alternate days to facilitate easy starting and smooth running in the case of the Cooper engine. In the Lister engine, the spray was quite good and there was no necessity to take the atomiser to pieces.

### **Valves**

There was neither pitting nor corrosion of the valves and the valves seats. The valves had a tendency to stick fairly frequently. However, there can be no doubt that with groundnut oil as fuel, sticking of the valves must be expected, due to its greater gumming capacity and therefore more frequent cleaning of the valves is required than with diesel oil.

### **Piston Wear**

The carbon deposit on the piston top was appreciable but it was soft and could be easily scraped on the Cooper engine. The piston wear varied

from 1/1000 in. to 3/1000 in. There was neither evidence of pitting nor corrosion in both the engines.

### Piston Rings

In the Cooper engine the first compression ring stuck in its groove at the end of 171 and 557 hrs. of run while the other rings were free. At the end of 858 hrs., the second compression ring was stuck. The piston rings were replaced.

### Cylinder Wear

The wear of the cylinder varied from 1/1000 in. to 1½/1000 in. There were no scoring marks on the cylinder walls. There was evidence of corrosion.

### Deposits

In the Cooper engine the exhaust pipe had to be cleaned thrice during the entire 1,000 hrs. run due to the clogging of the pipe with sticky carbon deposit. The excessive carbon deposit is due to the imperfect combustion in this engine and excessive lubricating oil consumed. This difficulty was not experienced during the 1,000 hrs. run on the Lister engine.

### Conclusion

The following observations may be made on the use of vegetable oils as fuel for slow and high speed diesel engines:

1. Compression ignition engines start easily from cold on groundnut oil. The atomizer must be kept free from carbon deposits. If any difficulty in starting is experienced, the engine may be started on diesel oil and changed over to groundnut oil after warming up.
2. Groundnut oil has no deleterious effects such as corrosion and pitting on various parts of the engine like cylinder, piston, valves, etc.
3. The maximum power output with groundnut oil is practically the same as with diesel oils.
4. The amount of carbon deposit is appreciably more with groundnut oil. This necessitates careful operation and strict maintenance routine.
5. Valves require more frequent cleaning and grinding due to their exhibiting a tendency to stick in their guides with groundnut oil as fuel.

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# Indigenous Fuels for Internal Combustion Engines

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## Introduction

Of the petrol substitutes, producer gas and alcohol have been used in India with some success. During World War II many motor vehicles were fitted with producer gas plants to overcome the shortage of petrol. Many difficulties were, however, experienced in overcoming the excessive engine wear. A number of testing stations for testing producer gas plants were set up throughout the country. The Master Testing Station was set up at Physical Laboratory of the Council of Scientific and Industrial Research, where research work to improve the quality of producer gas was carried out in addition to the routine testing of gas plants.

## Producer Gas

The fuel most commonly employed in India for producer gas generation is wood charcoal derived from forest wealth. It was feared that sulphur and nitrogen content of charcoal might lead to corrosion. However, recent tests<sup>1</sup> have indicated that the sulphur dioxide and ammonia contents of the gas from wood charcoal are very low and corrosion caused by them is not likely to be so excessive as to cause damage to the engine parts.

The dust content of the gas has been found to be a serious factor in the wear and tear of the cylinder and piston surfaces, unless special care is taken in filtration of gas and air. In view of the general belief that producer gas causes excessive wear, a new 3-ton Chevrolet truck fitted with a portable producer gas plant was used to determine the wear caused under Indian road conditions and it was concluded<sup>2</sup> that if proper attention is paid to gas filtration, the use of producer gas does not cause more wear than the use of neat petrol. The trouble, sometimes, is with the oil cleaner rather than with the fuel. It has been shown<sup>3</sup> that the conventional oil cleaners give varying degrees of effectiveness under varying atmospheric dust concentrations and engine speeds.

Realizing the importance of filtration, a number of investigations on the different aspects of filtration as applied to the problem of producer gas were carried out in this laboratory. The data collected from the studies on the flow of gas through fibrous materials<sup>4</sup> and textile fabrics<sup>5</sup> enables one to choose the material most suitable for use at the various stages of gas filtration. Suitable filter designs for use with these materials on a gas plant have also been suggested. A special type of comparatively light and fool proof filter, called the cyclone filter,<sup>6</sup> was developed. A later development<sup>7</sup> of a combina-

tion of a bubbling type of filter with two cyclone filters holds out promise of dispensing with the various types of dry filters commonly used at present.

The development of maximum power in producer gas engines depends to a large extent on precise control of the mixture strength. A gas carburettor<sup>8</sup> designed and constructed in the National Physical Laboratory has shown from 5 to 25 per cent extra power when compared with the usual carburettors depending upon the speed of the engine. The low calorific value of producer gas, the low volumetric efficiency of the engine due to high pressure drop across the producer and the high temperature of gas fed into the carburettor are the major considerations responsible for an appreciable reduction of power developed by the engine as compared with that obtainable with petrol. The maximum power with gas can be increased if the compression ratio of the engine is increased by suitable alterations to the pistons or the cylinder head or both. Spier and Giffen<sup>9</sup> found that operation at ratios higher than 10:1 was restricted by detonation. The greater part of the improvement in power output with increasing compression ratios was obtained below 10:1 compression ratio; the gain from ratios higher than this being small due to the marked increase in friction losses with increasing compression ratio. The petrol engines generally operating on producer gas have compression ratios ranging from 5 to 6.5.

Another method of obtaining increased power output is the use of supercharger operated by exhaust gas. The power output is then comparable with that of petrol. In the tests conducted by Brown Boveri,<sup>10</sup> the entire gas plant had to be made absolutely leak proof so as to avoid the dangers of carbon monoxide poisoning and the fire hazards involved by its use. However, this could be minimised by proper location of the supercharger.

### Power Alcohol

Because of low calorific value, the rate of consumption of alcohol is greater than that of petrol for equal power, although the higher latent heat of alcohol results in a lower suction temperature giving greater charge and greater maximum output at equal compression ratios. Higher compression ratios resulting in increased thermal efficiency are possible, because the addition of alcohol to petrol improves the anti-knock properties. It is, however, necessary to make a number of modifications in the petrol engine to make it suitable for use with neat alcohol. For example, the jet area in the carburettor has to be increased to secure comparable power.

Blends also affect fuel pump diaphragms, designed for use with neat petrol. The pump diaphragm consists of a linen base coated with chemicals insoluble in petrol but soluble in alcohol and alcohol-petrol blends. With the use of blends containing alcohol, the coating goes into solution and the efficiency of the fuel pump falls down and finally the pump stops functioning. The chemicals accumulate at the needle of the petrol jet carburettor. Experimental investigations on fuel pump diaphragms suitable for use with alcohol-petrol blends or neat alcohol have been made in this laboratory during the war

years. Long term trials at room and at low temperatures have proved the suitability of some of them.

### Diesel Oil Substitutes

Aggarwal, Chowdhury, Mukerji and Verman<sup>11</sup> demonstrated that most of the Indian vegetable oils could be successfully employed as diesel fuels. Some of them could be used without any modification to the engine, while others required the use of corrosion resistant alloys for certain engine parts. Cottonseed oil, in particular, gave exceptional performance in that its consumption was definitely less than that of mineral oils.

Despite good performance, vegetable oils have not been able to compete with mineral fuels, because of their higher price. In order that vegetable oils may be able to do so, colloidal fuels<sup>12</sup> containing low ash charcoal dispersed in groundnut oil were prepared. With the use of stabilizer made out of the oil itself, colloidal fuels containing as high as 35 per cent by weight of charcoal were prepared. These could be handled like liquid fuels and stored for long periods without the charcoal particles settling down. However, the injection system could not stand up to the abrasive action of the colloidal fuel. Further work towards reducing the wear of the injection system by using charcoal powder of 1 micron size or less, redesigning of the injection system and suitable modification to the combustion chamber is called for.

Last of all, it is worthwhile investigating the use of powdered solid fuels in internal combustion engines. Some experimental work<sup>13</sup> has already been done ; but it is necessary to carry out considerable amount of further research work to develop successfully the fuel as well as a suitable engine.

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# The Diesel Knock—Its Elimination in High Speed Diesel Engines

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To realise the advantages of the long established slow speed diesel in its high speed version, we must depend mainly on the speeding up of fuel combustion to about 10 times faster rate than in the slow speed engine. Such high speed combustion, however, approaches explosive ignition setting up pressure waves which spread with the velocity of sound and strike on the piston and engine parts, constituting hammer blow or 'Diesel Knock'. The effect of diesel knock makes engine running unsteady and gives rise to vibrations, shock loads and abnormal stresses that impair the life of the engine. Smooth running and the finer qualities of performance can be attained only by control of pressure rise so that a sustained pressure drives the piston instead of a hammer blow. Efforts made by various workers to eliminate the diesel knock and achieve efficient combustion in a short time with controlled pressure rise, constitute the history and development of the high speed diesel engine.

All the events from injection of fuel to maximum pressure rise happen during the short time interval required for the crankshaft to revolve through about  $30^\circ$ . In an engine running at 30 r.p.m. this represents  $1/60$  of a second, while at 1,500 r.p.m. it represents  $1/300$  of a second. Roughly half this period is utilized in a process of preparing the minute fuel droplets that have been injected into the combustion space with an initial velocity of nearly 40 ft. sec. to a condition of auto-ignition. This period is termed Ignition lag or Delay period. Once the particles of fuel first entering the combustion space have attained the condition of auto-ignition, further fuel particles which have already entered or continue to enter, do not require the preparatory period, and visible combustion spreads with flame propagation throughout the space exactly as in spark ignition petrol engines, but at considerably higher speed. It is in this phase that knocking occurs particularly when the delay period is long and fuel accumulation during the period is large. The maximum pressure attained by itself is not of significant effect as the engine parts can be designed to withstand any anticipated peak pressures ; but it is the high rate of rise of pressure that produces knocking and shock effects. The rate of rise of pressure is therefore a measure of diesel knock and can be well demarcated from the curve of rate of change of pressure. It is usual to state the maximum rate of rise of pressure per degree of crankshaft angle and this will measure the diesel knock characteristic of the engine. The earlier designs with open combustion chambers are typical of excessive

pressure rise rates—some designs giving as high a value as 40-60 lb. per degree of crank angle.

Improved designs of combustion chamber development have progressively attained smaller pressure rise rates by suitable combination of antechamber, swirl chamber, air cell, etc., so that combustion and rate of pressure rise are controlled without sacrifice of speed and efficiency. The ideal aim of controlled combustion will be to limit the maximum pressure to about the same as obtains in a petrol engine, combustion of fuel taking place at this sustained pressure, giving a high mean effective pressure but low maximum pressure. Fig. 1 shows the progressive improvement in pressure rise limitation, and one design—the Lanova—claims to have achieved combustion at constant pressure at a level of about 600 p.s.i. or 40 atm. by a combination of pre-combustion and vehement stimulated swirl effect in a specially shaped dual chamber. This engine has a compression ratio of 12.5 only.

### Combustion at High Speed versus Low Speed

Although our knowledge of combustion at high speeds approaching supersonic is far from complete, it has been established that the process is altogether different from the concept of combustion occurring at low speed as in the air-injection diesel where air injection of fuel takes place under more or less stagnant conditions in an open form of air chamber. Injection pressure near 4,000 lb./sq. in. give a velocity of around 600 ft. 1 sec. to the fuel droplets which under the high resistance from the stagnant dense hot air get widely dispersed and distributed in the combustion space. Heat transfer and chemical action prepare the first droplets into a state of self ignition in the delay period, after which inflammation of the fuel takes place in a progressive and sustained manner with the characteristics of combustion at constant pressure. The entire process is made up of (1) an endothermic stage when heat is absorbed by the fuel to prepare it for self ignition ; and (2) an exothermic stage where combustion of fuel evolves heat which energises the gases to exert pressure on the piston.

The first stage may be considered as a function of heat transfer under quiescent conditions with fuel particles having to seek out their requirement of oxygen for combustion, for which purpose they are supplied the requisite energy in the shape of high penetrative velocity. For a particular quality of fuel and conditions of pressure and temperature, the time required to attain the condition of auto-ignition is more or less constant and independent of the speed of the engine. This will apparently rule out any possibility of shortening the delay period to any marked extent. The attainment of efficient combustion even at speeds of 3,000 r.p.m. and above with the aid of stimulated turbulence, however, establishes that turbulence in some way alters the mechanism of combustion at high speeds and prepares the fuel to auto-ignition condition in a different way in a shorter time. The significant role of stimulated turbulence to obtain combustion at high speeds and with controlled pressure rise is now universally accepted and has served to promote new and improved

designs of high speed diesels. The theory that during the delay period fuel particles are first vapourised and then undergo self ignition cannot hold, since the process of gasification requires time which would render ignition at high speeds impossible of attainment. Further, the self ignition temperature of oil vapours is higher than those of liquid oils, and some of the constituents of the oil vapours have self ignition points higher than the compression temperature attained under the usual range of compression ratios adopted. Evidences accumulating indicate that auto-ignition occurs through a process of direct oxidation of hydrocarbon molecules, initiated as soon as fuel injection commences resulting in the formation of unstable peroxides which disintegrate with the evolution of heat and free oxygen. The process is continuously accelerated by stimulated turbulence and heat release from the unstable peroxide formation and disintegration, until visible ignition takes place, and this marks the end of the delay period. The duration of ignition lag under these conditions is thus controlled by the turbulence effect, which can be stimulated as a function of engine speed and thus renders high speed combustion possible.

The second phase of exothermic combustion of fuel in the low speed and high speed diesels also differs as regards proximity and distribution of fuel and oxygen molecules in the combustion space. The high penetrative velocity enables fuel particles to seek requisite oxygen in the quiescent and open combustion space of the former. In the latter, organized turbulence and a process of partial or dual combustion in separate spaces and the subsequent ejection of the partially burnt mixture of fuel and air through a suitably shaped passage in a continuous stream to the main combustion chambers enable progressive completion of combustion and sustained pressure rise, at high speeds. Thus in both phases turbulence provides the means for speeding up ignition lag and regulating combustion to eliminate detrimental pressure rise that may lead to knock effects.

Some experiences and investigations on ignition lag and pressure rise during combustion in an experimental engine are recorded in the diagrams in figs. 1A and 1B. The following are the particulars of the engine:

#### SINGLE CYLINDER ENGINE

Bore	...	...	...	...	100 mm.
Stroke	...	...	...	...	130 mm.
Displacement	...	...	...	...	10-20 c.c.
Compression ratio	...	...	...	...	14
Compression pressure	...	...	...	...	470 p.s.i.
Fuel Injection Pressure	...	...	...	...	2,500 p.s.i.

The working details of the engine are given in Table 1.

TABLE 1—WORKING DETAILS OF THE ENGINE

			Test 1	Test 6
Fuel injection				
Begins at	...	...	-23°	-23°
Ends at	...	...	-10°	+12°
Ignition lag				
In air cell	...	...	8°	12°
In engine cylinder	...	...	13°	17°
Maximum pressure				
In air cell	...	...	545 p.s.i.	870 p.s.i.
In engine cylinder	...	...	610 p.s.i.	690 p.s.i.
Load	...	...	nil	4.65 h.p.
Speed	...	...	980 r.p.m.	1,000 r.p.m.

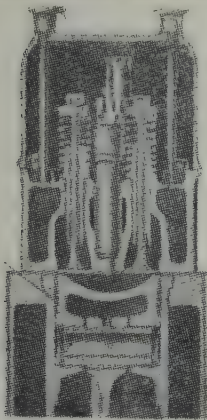
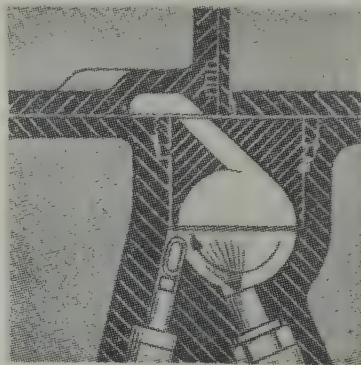
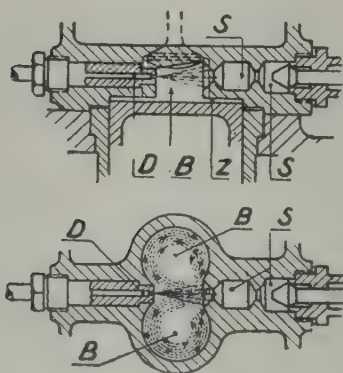
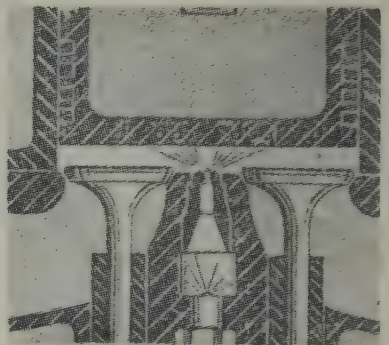
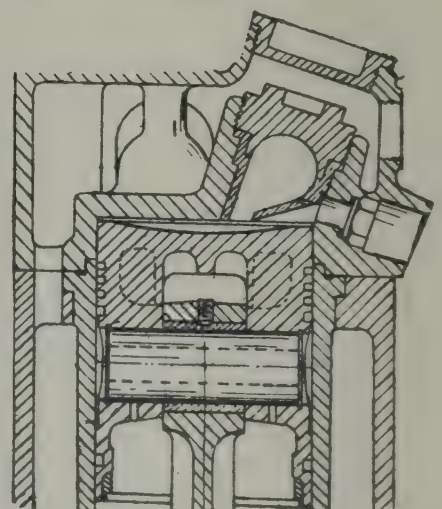
OPEN COMBUSTION  
CHAMBERANTECHAMBER &  
SWIRL CHAMBER'LANOVA' COMBUSTION  
CHAMBERSINGLE CYLINDER  
ENGINE

FIG. 1—SINGLE CYLINDER ENGINE AND THE CHAMBERS

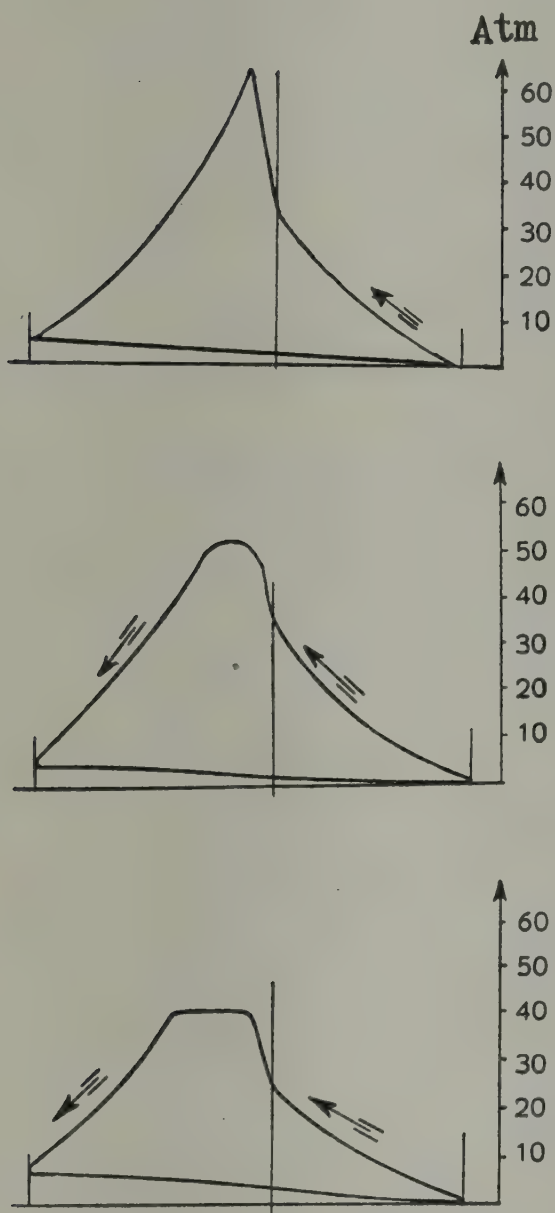


FIG. 1A—IGNITION LAG DURING COMBUSTION

The combustion chamber employed is a combination of a swirl chamber or air cell and open chamber connected by a passage. In both the full load and no load diagrams, the effect of swirl and turbulence on the ignition lag and the pressure rise in both the cell and open chamber are observable. The delay

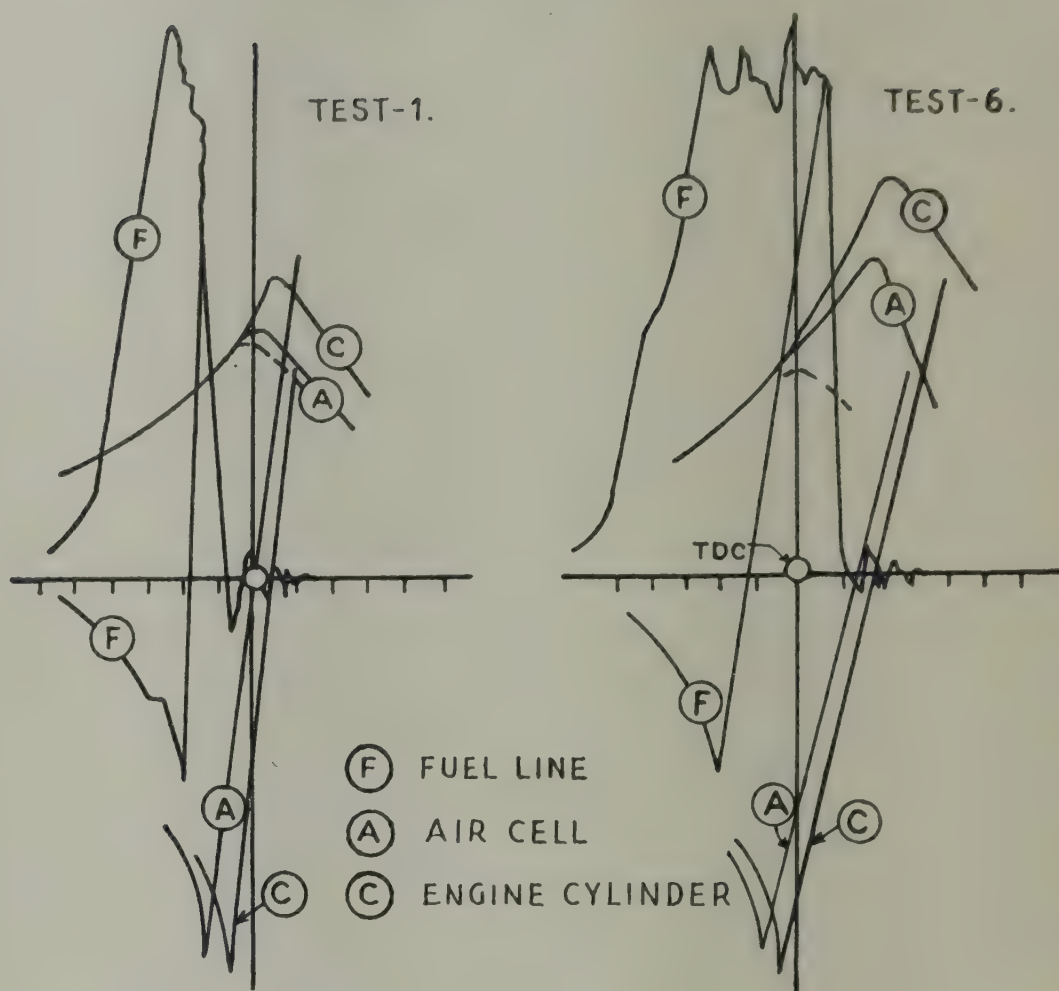


FIG. 1B—PRESSURE RISE DURING COMBUSTION

period is seen to be shorter for combustion in the swirl chamber than in the open chamber although fuel enters both simultaneously. The maximum pressure in the open chamber is also higher and occurs after the attainment of maximum pressure in the cell, indicating a dual process of combustion.

# Some Fundamental Aspects of Combustion and their Application in Internal Combustion Engines

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## Introduction

There exists hardly any other province in Internal Combustion Engineering in which fundamental research is so closely linked with practical application in the engine, as the combustion process. The details of the combustion process are characteristic for every fuel and determine how the engine should be designed to make use of a particular fuel. The combustion process is of decisive value for the thermal efficiency of the power plant, and hence for the fuel consumption, and affects directly the power of the engine by determining the effective pressure and limiting the number of power strokes in unit time.

Considering, for example, the reciprocating petrol engine, its economy and output are governed by the tendency of the fuel-air mixture towards knocking at a certain compression ratio. The tendency towards knocking is fundamentally a property of the fuel, but is influenced in a complex manner by engine conditions. The problem of increasing the compression ratio thus presents two sides for an attack: the purely chemical approach to influence, by additions, the reactions of combustion and what might be termed the physical approach of influencing external conditions during combustion.

Discussion here is mainly on the physical effects of external conditions on combustion.

## I. Modes of Flame Propagation

(a) *The flame of the Bunsen burner*—The velocity of a flame can be measured by observing the shape of the flame front of a Bunsen burner and by measuring the mass flow of the gas. These methods are the results of researches which have become classical,<sup>1</sup> but they give some of the principal laws governing the rate of flame propagation. The normal velocity of propagation of the flame could be observed in a plane flame front which is at right angles to the direction of gas motion uniformly distributed over the cross section. The plane flame front, however, is not stable, and any inclination between the flame front and the direction of the gas flow leads to a flame velocity which is greater than the normal velocity of flame propagation. This accounts for the increase in the area of the flame front when rapid combustion is desired. One of the causes for spreading out the flame front is the occurrence of turbulent motion in the gas flow, and the same effect can be established by vorticity created at the boundaries of the

flow. As a consequence of the vorticity in the gas the surface of the flame is increased with resulting increase in the rate of combustion.

The order of normal flame velocities occurring in Bunsen burners is roughly between 1 and 10 ft./sec. It has a lower limit on the lean side. The lower limit of inflammability is rather close to correct mixture.

G. Damkohler<sup>2</sup> has undertaken experiments on propane flames in Bunsen burners, and has shown that this effect is due to a type of turbulence where the average dimensions of disturbances and of exchange processes in the flow are large as compared to the thickness of the flame front, encountered in the laminar case: iniquities of the flame front are visual and the rate of combustion related to the cross sectional area is increased. As compared to this, any other type of turbulence provides for exchange on a much smaller scale which enhances the rate of the microscopic processes of diffusion in the flame front itself, thus resulting in an increase of the rate of combustion without visually disturbing the flame front.

A similar explanation may account for the increase in flame velocity if the burner is exposed to sound waves. These investigations have been carried out by H. Hahnemann and L. Ehret.<sup>3</sup>

The speed of the flame also depends on the rate of heat transfer, away from the locality of the flame. These influences may often be small, but will play an important part where the ratio of surface to volume of the combustible mixture is large.

In general, inflammability of a combustible mixture depends largely on the air-fuel ratio which has a rich and a lean limit and, on the leaner side, the speed of the flame is considerably reduced as compared to the speed of the stoichiometric mixture. Increasing pressure and temperature extend the limits of inflammability.

(b) *The flame in open tubes*—The actual velocity of propagation of the flame in the Bunsen burner was defined by the equilibrium between gas and flame velocity. In a tube open at one end and filled with a combustible mixture which is ignited by a spark located near the open end, the flame will travel from there towards the closed end and the exhaust gases leave the tube at a high velocity without essentially influencing the pace of the flame front. Thus the pressure in the fresh gas which remains at rest throughout will be more or less constant. The flame velocity can be determined if the flame is photographed on a film moving with a known velocity.

This method of determining the true flame velocity has been used in classical experiments, but on closer observation it can be seen that even in this case a plane flame front cannot develop. In fact, the shape resembles somewhat a hemisphere, which, however, will be disturbed by buoyant movements, the effect of gravity, and influence of the wall.

These effects have become clear mainly from Schlieren protographs taken with high frequency spark illumination, which have been published by U. Neubert,<sup>4</sup> who assumes that at a critical Reynolds number the flame profile turns from a fairly well defined shape into a distorted shape. This shape becomes disturbed by primary turbulent motion which may be superimposed

by turbulence created at the wall. A photograph from Neubert's publication is given in Fig. 1.

A new phenomenon which was not encountered in the Bunsen burner

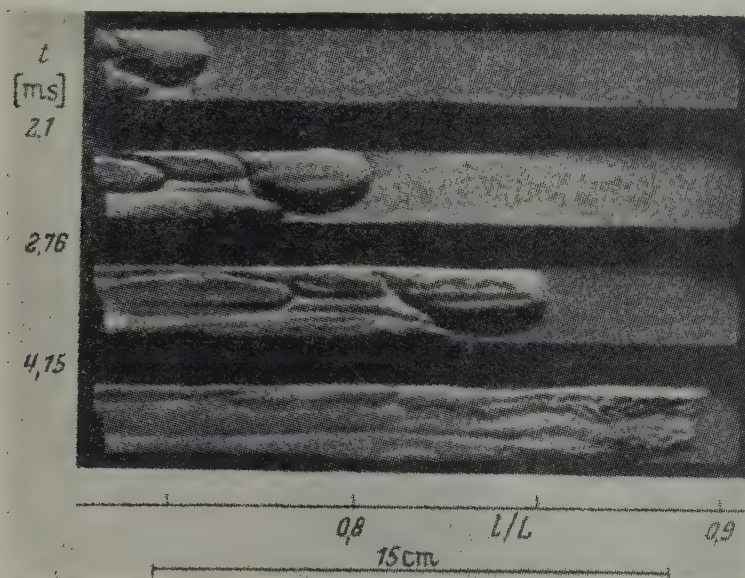


FIG. 1—PROPAGATION OF A FLAME IN A VESSEL WITH RECTANGULAR CROSS SECTION. INCIPIENT TURBULENCE. Schlierenphotograph (Neubert). (MS = milli-seconds)

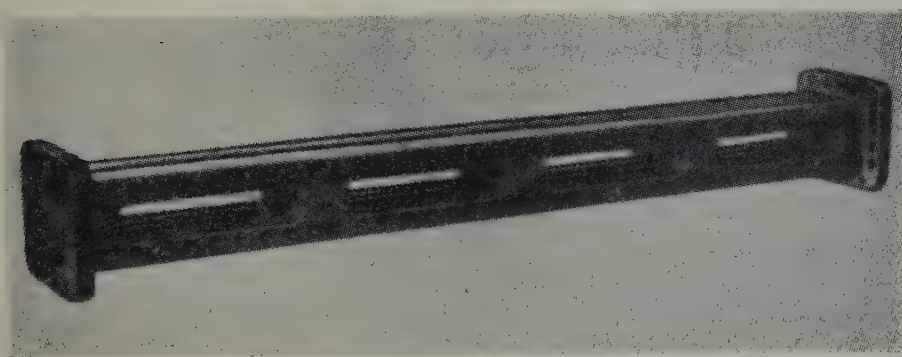


FIG. 2—COMBUSTION VESSEL WITH GLASS WINDOWS

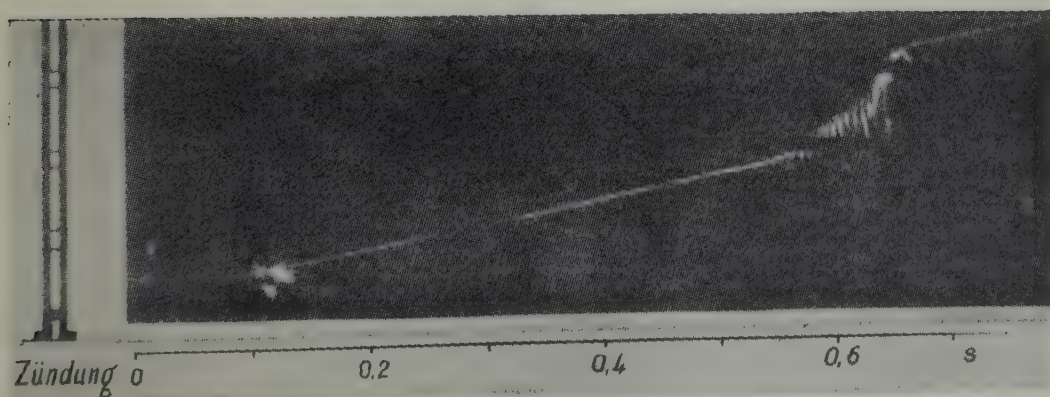


FIG. 3—COMBUSTION, PHOTOGRAPHED IN ITS OWN LIGHT, OF A PROPANE—AIR MIXTURE AIR RATIO = 0.8, OPEN VESSEL, IGNITION AT THE OPEN END. Horizontal vessel (Steinicke). Zündung = Ignition ; s = seconds

flame is often met in open combustion vessels (Fig. 2). The flame oscillates at some distance from the point of ignition (Figs. 3 and 4). Steinicke<sup>2</sup> considers this to be due possibly to periodic separation of eddies from the wall which excite the flame to oscillate. For larger amplitudes the effect of the oscillations should be an increase in the flame velocity, but no exact information is available. It has, however, been observed that the oscillating flame accelerates itself so that finally normal combustion may turn into detonation.

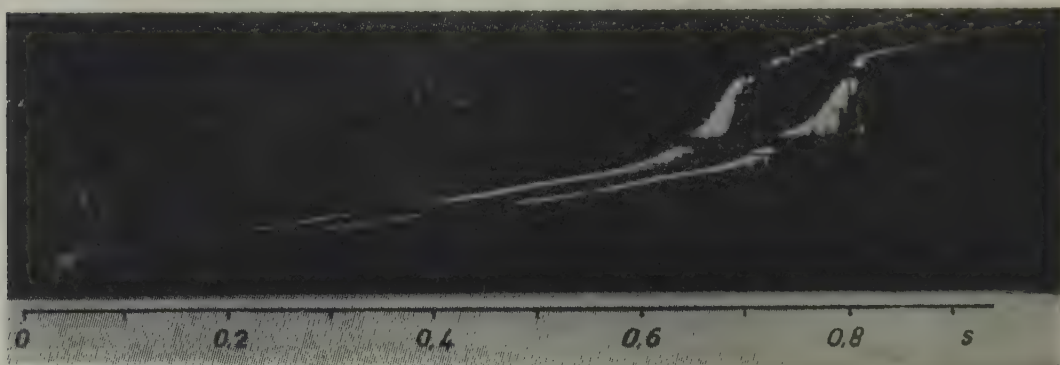


FIG. 4—COMBUSTION, PHOTOGRAPHED IN ITS OWN LIGHT, OF A PROPANE—AIR MIXTURE, AIR RATIO = 0.8, OPEN VESSEL, IGNITION AT THE OPEN END. Vertical vessel (Steinicke)  
s = seconds

In detonative combustion the flame velocity is extremely high and attains values of several thousand meters/sec. The flame in detonation is coupled to a pressure wave which ignites the mixture on passing.

For the creation of a pressure wave, a small initial compression may be assumed either due to the motion of a piston or due to the expansion of a small volume of gas, say at the closed end of a tube. The pressure amplitude may be small, but will proceed through the gas with a velocity slightly greater than the velocity of sound. On further compression by the piston, a new pressure wave is created, which now moves faster than the first owing to the higher temperature of the gas which was compressed by the first compression. On continuous motion of the piston, or any other displacing action, a system of pressure waves is set up, which ultimately superimpose one another to form a very steep pressure wave, which may ultimately cause self ignition in the gas, coupling the pressure wave with the reaction of ignition in the form of "detonative combustion" (Fig. 5). Pressures in the flame front may amount to 40 kg./cm.<sup>2</sup>

Under normal circumstances a considerable distance is necessary for the development of detonation, and it is normally restricted to oxygen-fuel mixtures. In very long tubes, however, air-fuel mixtures also may ultimately detonate.

If the propagation of flames in vessels, open at one end, is considered, it is important to state where ignition occurs. If ignition occurs at the open end, the burnt gases escape freely. If, however, combustion is initiated at the closed end, the expanding gases carry the flame front ahead. There the pressure will remain practically constant. This results in an increase of the

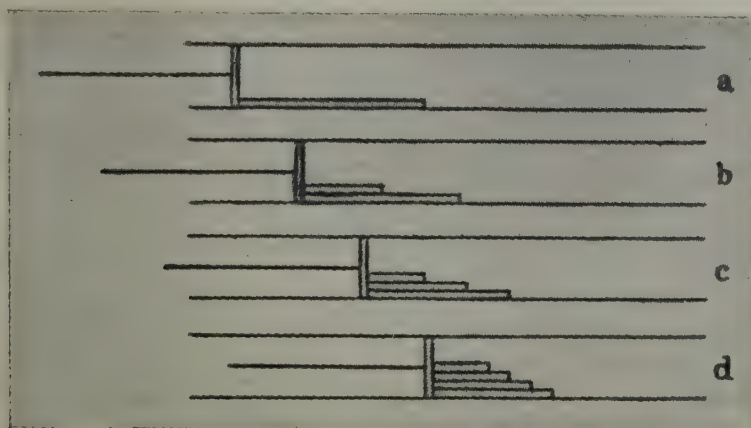


FIG. 5—ESTABLISHMENT OF A SHOCK WAVE IN A GAS  
(Becker)

visual flame velocity and is of special importance, if flames in closed vessels are considered.

(c) *The flame in closed tubes*—In closed vessels this phenomenon is characterised by the strong influence of motion of the gas. A partial volume of gas, when ignited, will expand and compress the remaining unburnt portion. The flame front will therefore move as a result of the expansion or contraction of the gases and the relative motion between gas and flame.

In closed vessels oscillation of flames has been observed quite early and similar explanations as in the case of open vessels have been given, but the processes are still not quite understood.

As the flame proceeds the overall pressure in the vessel will increase steadily, and this will ultimately lead to a lowering of the flame velocity, so

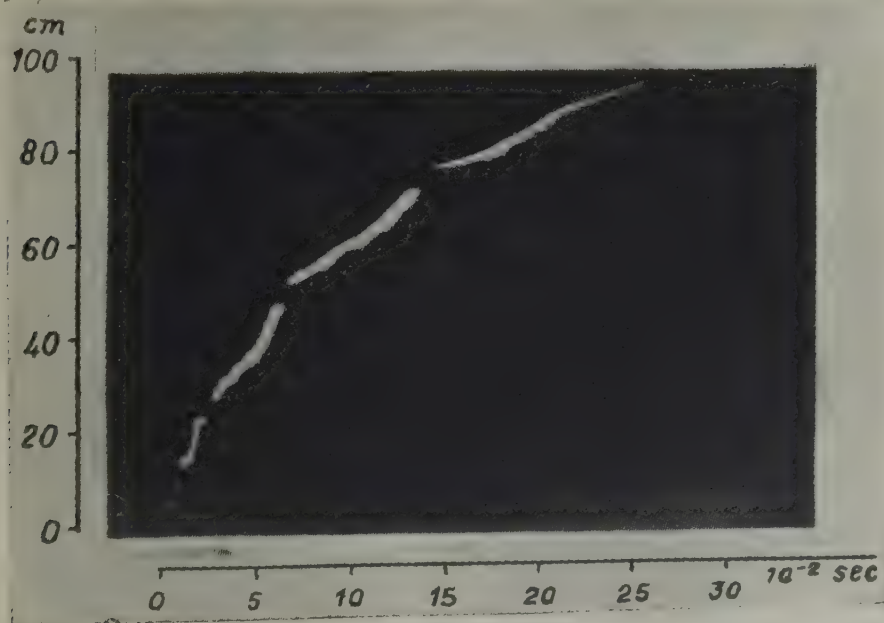


FIG. 6—COMBUSTION, PHOTOGRAPHED IN ITS OWN LIGHT, OF A  
CHEMICALLY CORRECT PROPANE—AIR MIXTURE, CLOSED VESSEL  
(Steinicke)

that after an initial rapid progress of the flame it will slow down on reaching the closed end of the vessel (Figs. 6 and 7).

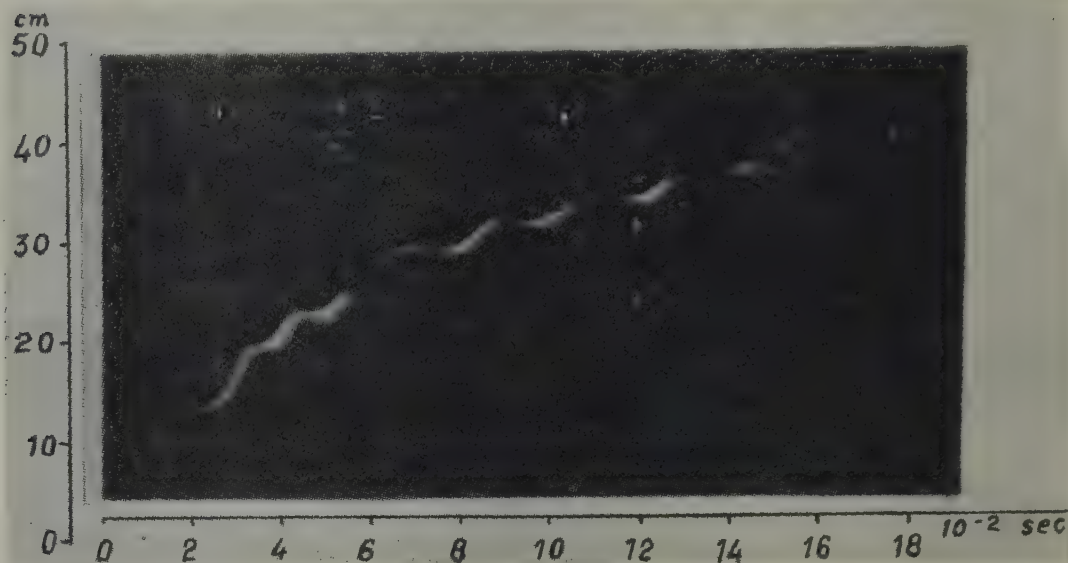


FIG. 7—OSCILLATIONS IN THE COMBUSTION PROCESS OF A PROPANE—AIR MIXTURE, AIR RATIO=0.6, PHOTOGRAPHED IN ITS OWN LIGHT. CLOSED VESSEL (STEINICKE)

Damkohler<sup>7</sup> has theoretically investigated the motion of gas in a closed vessel, and actual photographs have confirmed his findings. In the initial stages rapid expansion occurs so that for e.g., when one half of the weight of the charge is burnt it occupies nearly 80 per cent of the volume (Fig. 8). The unburnt gases, consequently, will be compressed to less than half of their original volume, and on the assumption of adiabatic compression, the temperature will have been raised, so that, under certain circumstances, instantaneous self ignition may occur in the rest of the charge, especially since the temperature in the remaining volume is uniformly distributed. This results in a very high pressure as soon as self ignition occurs which by far exceeds the pressure in the rest of the combustion space.

This pressure difference, created by self ignition, will, of course, exist only for a very short period, and immediately a shock wave will tend to equalise the pressure in the vessel. This process will create oscillations of high pressure amplitudes which will pass through the combustion space and will be reflected on the walls, thus creating a complex and continuously changing field of pressures. The mechanics of pressure waves will naturally have a large bearing on the details of this field. There is every indication that the rate of heat transfer will be greatly increased so that a substantial part of the internal energy of the gas is dissipated through the walls.

The outstanding point in the mechanics of the combustion process in closed spaces, is the condition of the unburnt gas leading possibly to self ignition, and the dissipation of energy released in this partial combustion into the previously burnt gases, together with the phenomena accompanying the equalisation of the pressure differences set up in the gas space.

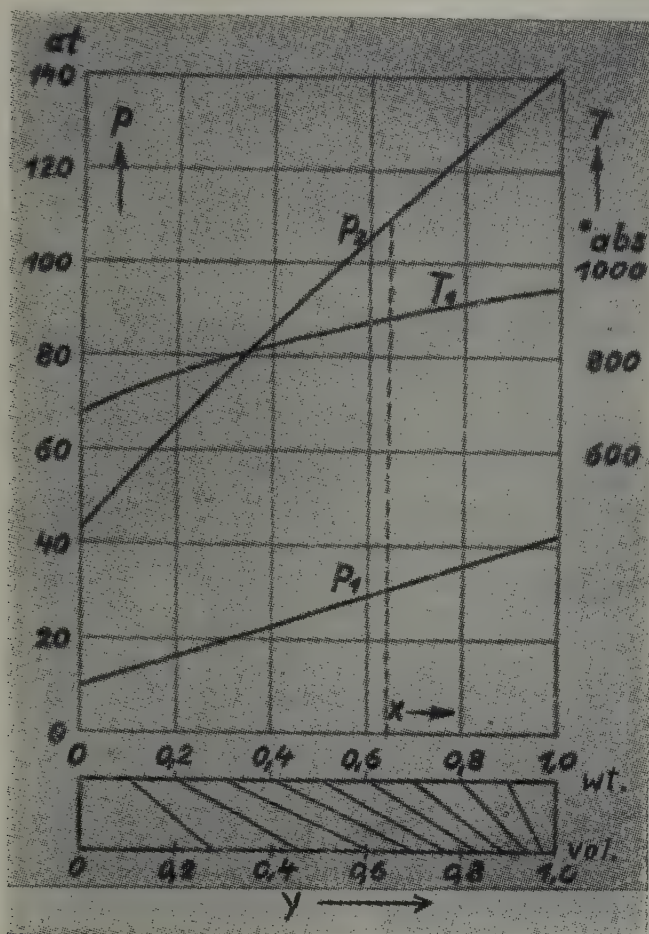


FIG. 8—PRESSURE  $P_1$  AND TEMP.  $T_1$  IN THE END MIXTURE IF FRACTION  $x$  OF WEIGHT & FRACTION  $y$  OF VOLUME IS BURNT. IF SELF IGNITION OCCURS PRESSURE WILL BE  $P_2$  (DAMKOEHLER).

Self ignition may or may not occur, depending on the chemical nature and physical influences acting on the highly compressed and heated unburnt gas. A considerable amount of research has therefore been concentrated on studying the details of self ignition of a compressed the heated combustible gas.

## 2. Motoric Combustion

(a) *Specific problems of motoric combustion*—The specific problems encountered in motoric combustion, which are difficult to be established in those laboratory set-ups as discussed so far, are essentially connected with the conditions of the charge in the engine: the charge itself has imposed rapid movements with turbulence as a consequence of the intake process. Both the absolute velocity of the combustible mixture and its turbulent state account for the high velocity of propagation in the engine, as compared to the normal flame velocity.

Another specific feature of motoric combustion is “after burning”. This effect may be due to recompression of this part of the charge due to the expansion of the gas in other portions of the combustion space. On the other

hand so-called pre-reactions<sup>8</sup> take place previous to the actual combustion, with the exception of benzene and alcohol, and they can be assumed to be assisted by high coolant and intake temperatures, and suppressed if the engine speed is high. By far the most important occurrence of motoric combustion is knocking which becomes understandable after what has been said above. The extremely rapid and spontaneous combustion of the last precompressed part of the charge causes pressure waves in the combustion space, which are audible, and which by a considerable increase in the rate of heat transfer decrease the mechanical output of the engine. Besides, heavy knocking will ultimately lead to the destruction of the engine.

The on-set of rapid combustion is not linked solely with the attainment of the self ignition temperature, but there is a time lag before a flame occurs. This time interval also depends on the mixture strength of the charge, the details of the combustion space geometry and the conditions therein, namely, the pressure and temperature, or rather, the temperature history, and is different for different fuels. Possibly therefore, whereas all outer conditions are given for self ignition in the end mixture this may not occur due to lack of time—i.e., the flame front consumes the end mixture before it finds time to ignite itself. For higher speeds therefore, the tendency towards knocking will be lessened.

(b) *Fundamental researches simulating motoric conditions*—The behaviour of the end mixture of the charge in the engine is thus primarily dependent on the self ignition temperature. The temperature is the result of the conditions of the surroundings such as the rate of heat transfer to the walls. If chemical energy is liberated faster than dissipated to the walls the temperature will rise and ultimately reach the self ignition temperature. On the other hand, if heat is dissipated at a considerable rate, self ignition may not occur. Further, the magnitude of ignition delay will ultimately decide whether knocking may or may not occur.

A number of authors<sup>9</sup> have developed special laboratory apparatus working mostly with a piston moved by the impact of a weight or with compressed air. Initially, the piston speed was kept small, so that pre-reactions and heat transfer had a noticeable effect and the piston was allowed to travel back after compression. In recent times it was managed to keep the piston at the upper dead centre of its stroke, or prevent the compressed charge from expanding by other means.

H. A. Havemann<sup>10</sup> designed an apparatus shown in principle in Fig. 9 for the rapid compression of a combustible gas mixture, with a view to study the effect of gas oscillations (Fig. 10) on heat transfer (Fig. 11) and flame propagation (Fig. 12). Both quantities are generally increased with the intensity of pressure fluctuations. Details of ignition lag, influence of pressure, temperature and mixture characteristics and the effect of additions to suppress knocking were investigated by F. A. F. Schmidt<sup>11</sup> (Fig. 13) and lately by R. C. Spencer.<sup>12</sup> Results in general indicate that ignition lag decreases with increasing pressure and temperature, and increases on the rich

and lean side of the mixture strength, and can be represented, for not too large a temperature interval, by a formulae of the form:

$$\text{Ignition} = \alpha \cdot \beta \cdot \frac{e^{b/t}}{h^n}$$

where  $\alpha$  is dependent on the fuel characteristics,  $\beta$  denotes the influence of temperature increase during the period of ignition delay and  $b$ ,  $n$  define values to be found from experiments.

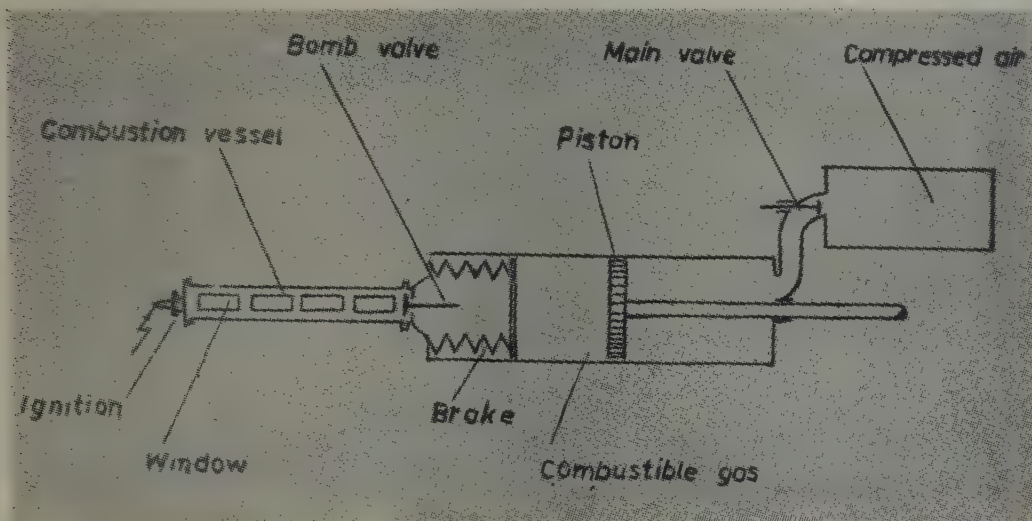


FIG. 9—PRINCIPLE OF THE APPARATUS FOR RAPID COMPRESSION OF GASES IN VESSELS WITH GLASS WINDOWS

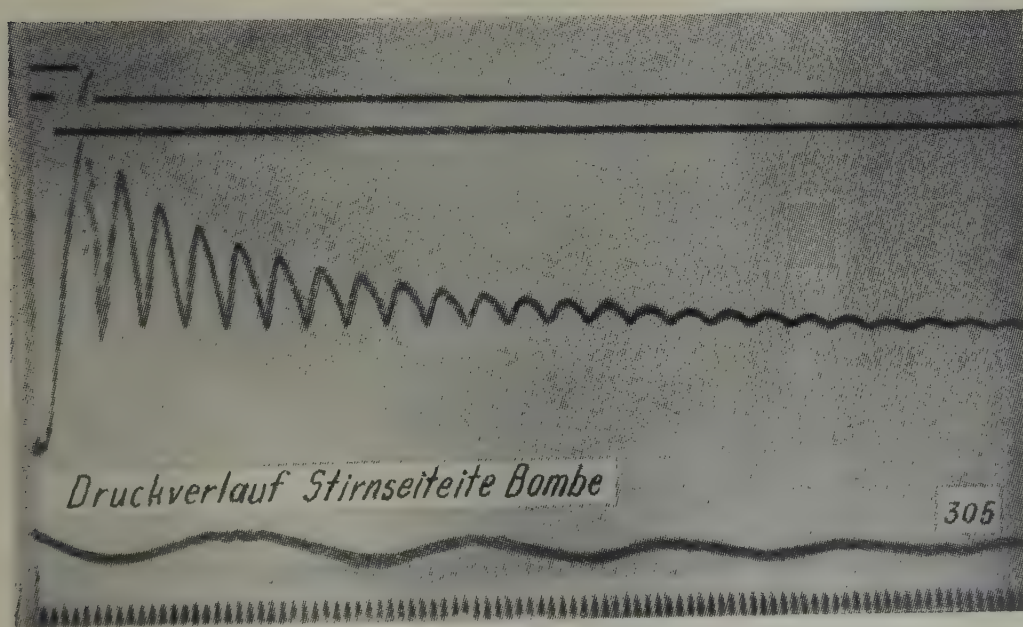


FIG. 10—PRESSURE IN THE COMBUSTION VESSEL, OF RAPIDLY COMPRESSED AIR AFTER COMPRESSION

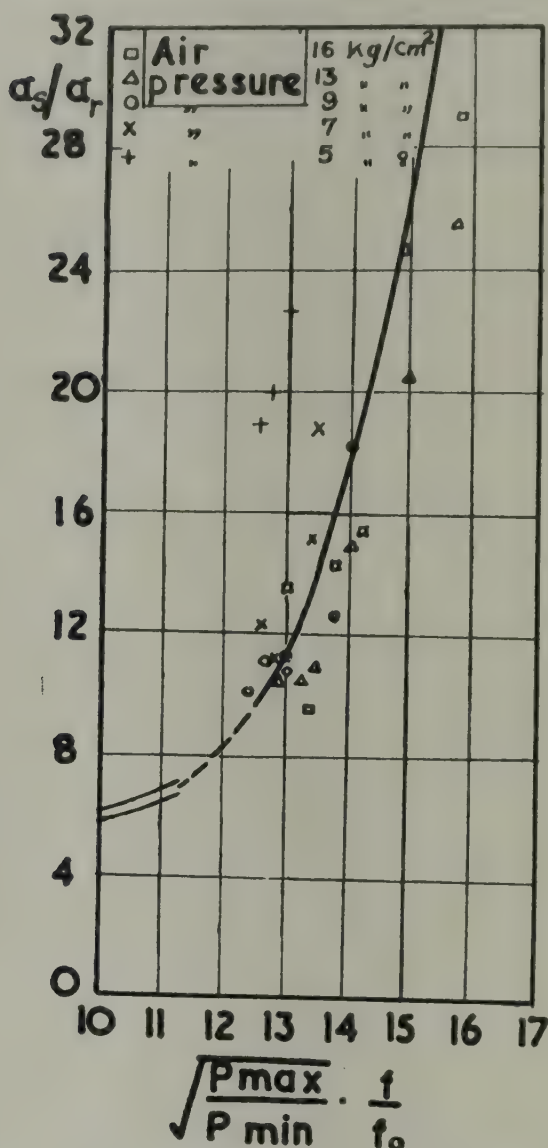


FIG. 11—RATIO OF HEAT TRANSFER COEFFICIENTS FOR OSCILLATING AND STILL GAS, DEPENDENT ON PRESSURE AMPLITUDE AND FREQUENCY OF THE PRESSURE OSCILLATIONS

$\alpha_s$  = HEAT TRANSFER OF OSCILLATING GAS

$\alpha_r$  = HEAT TRANSFER OF STILL GAS

It must, however, be mentioned that the behaviour of some fuels cannot be predicted. This may also apply to blends of commonly used fuels with unorthodox fuels.

A different method was introduced lately: Fuel was injected into a pressurized and heated vessel,<sup>13</sup> and the time for ignition measured. The results, if differences due to the time necessary for evaporation are eliminated, conform to the results obtained in the compression apparatus.

(c) *Investigations under motoric conditions*—The technique of investigations is concerned mostly with the chemical analysis of the combustion process, say, by sampling gas probes from the cylinder of the running engine, or by

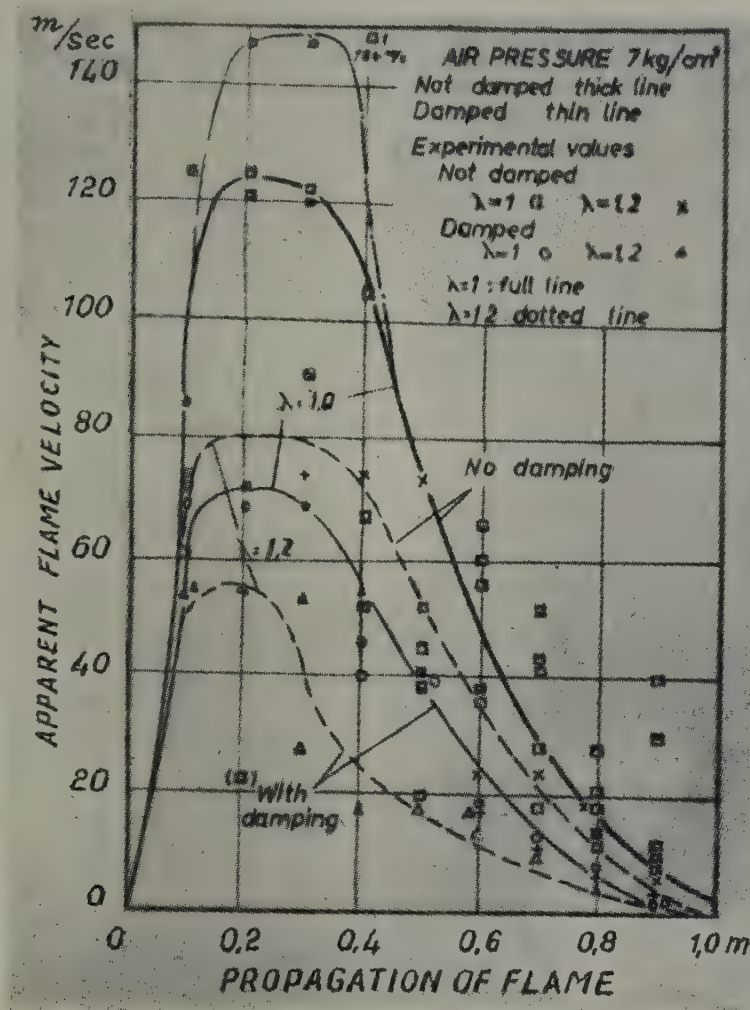


FIG. 12—EFFECT OF GAS OSCILLATIONS ON APPARENT FLAME VELOCITY, FOR A PROPANE AIR MIXTURE, AIR RATIO 1:2

photographic examination of the process of combustion through quartz windows. The Schlieren method and the method of ionisation on several points in the combustion space have also been employed to record the passing of the flame.

The time available for the combustion of the charge is in the order of less than  $3 \times 10^{-3}$  sec., and the actual flame velocity is therefore about 10 times larger than the normal velocity. This increase is possible due to vorticity and turbulence of the charge. In an engine, therefore, all such means will have an influence, which react on the vorticity in the charge, such as the engine speed.

The influence of mixture strength on flame velocity is found to be in conformity with findings in bomb tests. The velocity has a maximum on the rich side of the correct mixture strength, at an air ratio of 0.85 in the average. On the lean side, the velocity is smaller and uneven running is likely to be encountered. Besides, the combustion process is prolonged and a decrease in

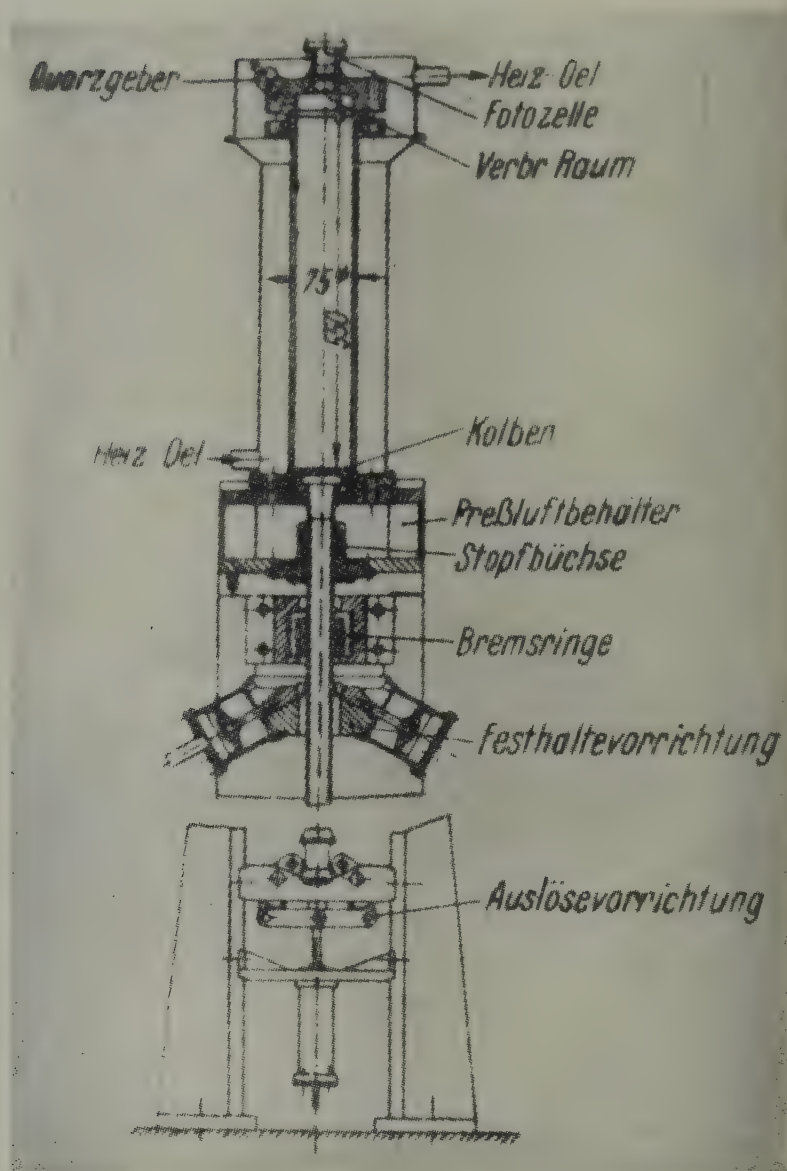


FIG. 13—APPARATUS FOR QUASI-ADIABATIC COMPRESSION OF FUEL VAPOUR AIR MIXTURES

efficiency is unavoidable. The tendency towards knocking is most pronounced at a mixture strength near to stoichiometric.

The examination of knocking combustion has contributed greatly to ascertain the mode of flame propagation in this case. It could be proved photographically<sup>14</sup> that the end mixture is ignited ahead of the normal flame front. The conditions in the mixture have a great influence. Increasing the compression ratio has the greatest effect, and an increase in temperature and pressure enhances the tendency towards knocking, and a similar influence is exerted by the mixture strength (Fig. 14). All these factors affect directly the reaction in the end mixture. The pressure fluctuations in the cylinder (Fig. 15) explain the rapid rate of reaction and the increase in heat transfer.

The original idea behind early investigations that detonation may occur in the engine, could not be proved ; detonative combustion requires a greater

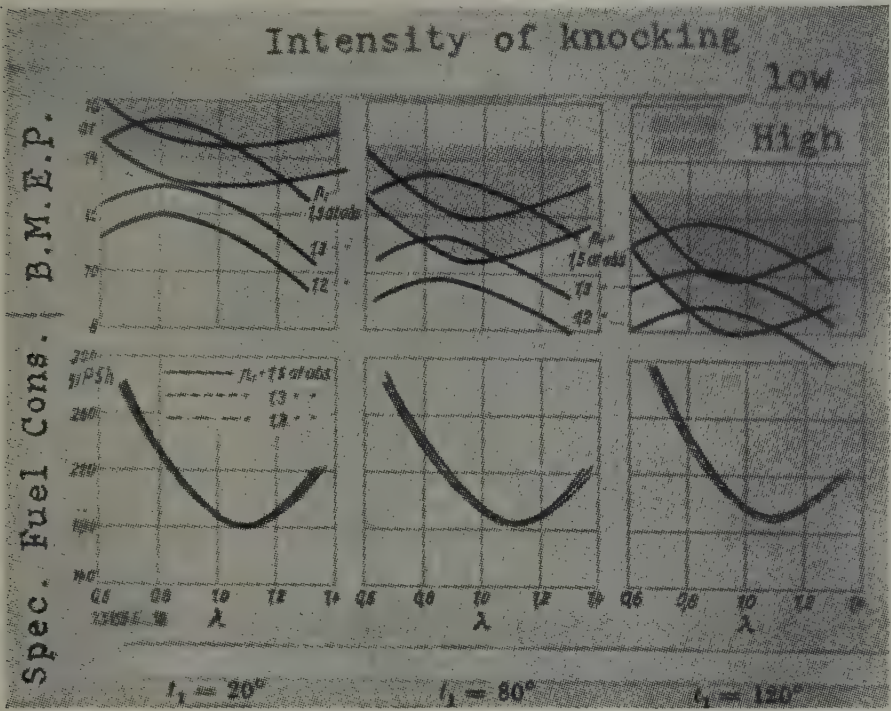


FIG. 14—MEAN EFFECTIVE PRESSURE, SPECIFIC FUEL CONSUMPTION & KNOCK INTENSITY DEPENDING ON MANIFOLD PRESSURE & TEMPERATURE (F.A.F. SCHMIDT)

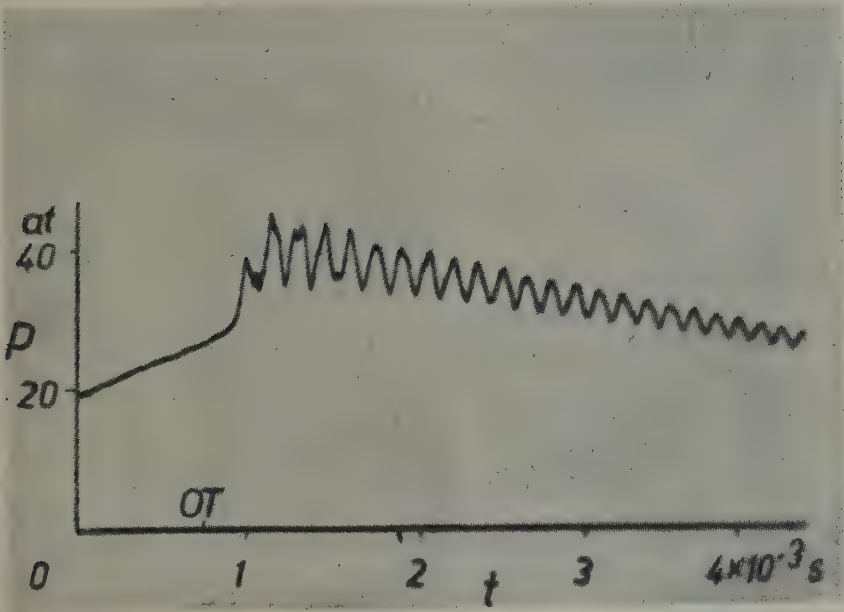


FIG. 15—PRESSURE DIAGRAM FOR KNOCKING COMBUSTION (GOHLKE)  
P = PRESSURE

length of the combustion space to develop fully. The velocity of detonative combustion, further more, is rather independent of the nature of the fuel—which is not the case for knocking combustion, and moreover certain additions, which prevent or defer knocking, have no influence on detonation.

### 3. Application of Fundamental Researches

(a) *The elimination of knocking combustion*—There are two rather distinct schools of thought aiming at the prevention of knocking. One school applies chemical agents which bring about chain reactions and thus influence the process of reaction. The other restricts itself to physical methods which influence the condition of the end mixture so that knocking is repressed. A more recent development tries to solve the problem by preventing an end mixture being formed.

How external factors can influence knocking combustion are discussed below leaving chemical methods aside.

(i) At first,<sup>7</sup> it may be assumed that the mixture is uniformly distributed in the combustion space and as such the flame velocity relative to the gas will be essentially constant, but movement of the charge will lead to local disparities in the rate of flame propagation. For this case, the geometry of the combustion space, the location of valves, and of the spark plug have a marked influence, besides such factors as the nature of the surface and the material of construction. It is assumed here that carburettor is used and the fuel is completely vaporized.

The role played by the geometry of the combustion space in the case of detonative combustion has a parallel in the case of knocking combustion. From the foregoing it can be seen that the stretched combustion space will favour knocking while a compact shape will be detrimental to incipient knocking. Hot spots, especially if they are in the region where the end mixture is formed will result in knocking since the point where the end mixture is formed will depend greatly on the location of ignition. The location of the spark plug is very important. In general, it should be opposite to the best cooled regions and in the vicinity of any hot spots, especially the exhaust valve.

All measures to limit the temperature of the end mixture will thus be beneficial and this can be achieved by cooling effectively those parts of the combustion space into which the flame is proceeding. Also some turbulence will lead to a high flame velocity so that the ignition lag in the end mixture will be too great to allow knocking. As such a combustion chamber built so that the piston comes very close to the head at the end of the compression stroke results in high turbulence and in intensified heat transfer and compression ratios of 12.5:1 have been applied without knocking in engines applying this design principle.

(ii) It may now be intended to use the disparity in the tendency towards knocking of different mixtures so as to prevent knocking (Fig. 14). Factors mentioned above may additionally assist in this endeavour. In the case of a carburettor, stratification of the charge is extremely complicated. Injection of fuel may offer a better solution if the following could be achieved in principle.

A mixture conducive to high flame velocity should be accumulated around the spark plug so as to ensure ignition. With increasing distance from the point of ignition the mixture should become leaner, so that, at the far end of the combustion space, where knocking conditions may eventually be

established, the mixture is too lean for self ignition. Besides, for lean mixtures ignition lag would be large so that the end mixture could be consumed before knocking occurs. This arrangement would allow an appreciable increase in compression ratio thus increasing also fuel economy.

Fuel injection in general would offer other advantages, such as an unrestricted induction passage for air, equal distribution of fuel for individual cylinders, guaranteed atomization, greater density of the charge, and thus increased output, diminished danger of icing, etc. Ultimately as scavenging with air becomes possible, the two-stroke engine would have every hope of overcoming the high fuel consumption and many problems of the highly supercharged aero engine would be solved. It must, however, be mentioned that scavenging with air and a large valve overlap increase the tendency towards knocking. The dilution of the fresh charge with unburnt gases represses knocking to a large extent, and thus complete scavenging will not be advisable.

The stratification of the charge as described can be obtained by injecting the fuel directly into the cylinder, and tests have shown that the air ratio, normally restricted to values of 1.15 to 1.25, can be extended to 1.4 to 1.5, and thus would reach values hitherto applied only in diesel engines. A similar effect is established if injection is undertaken in two steps, the first to create a uniform and lean mixture, the second to secure the enrichment of certain portions in the combustion space to ensure good ignitability of the charge there.

(iii) The formation of the end mixture is prevented if fuel is injected at the rate it is burnt and consumed. This, in practice, means that a regular flow of air is created into which fuel is injected progressively at a predetermined rate. In the Texaco system,<sup>15</sup> the swirl created in the combustion chamber moves the fuel towards the spark plug (Fig. 16). The duration of injection

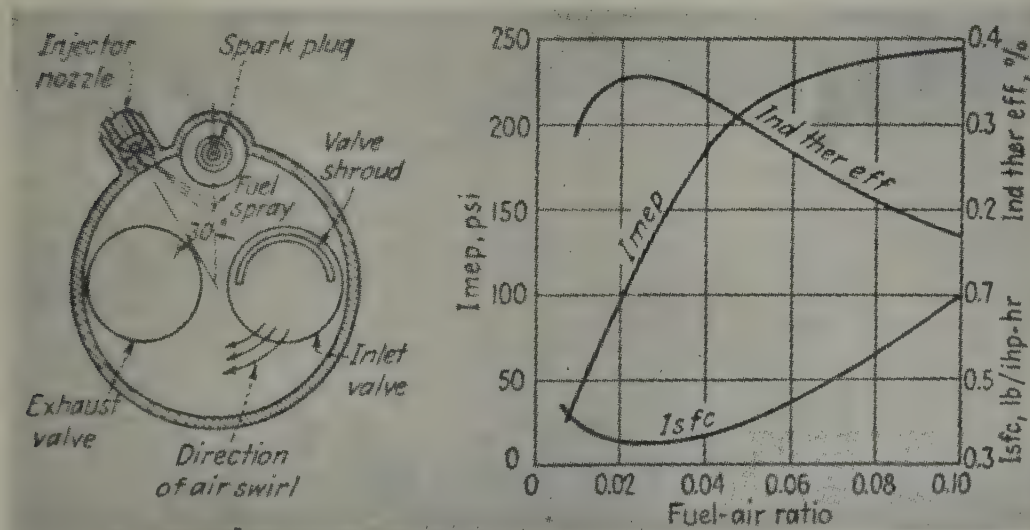


FIG. 16—KNOCK-FREE COMBUSTION CHAMBER AND PERFORMANCE CHARACTERISTICS WITH KNOCK-FREE COMBUSTION CHAMBER. ( $R=10:1$ . INTAKE-MANIFOLD PRESSURE 60 IN. HG: ENGINE SPEED 1,800 R.P.M., UNCORRECTED FOR WORK OF SUPERCHARGING (BARBER, MALIN AND MIKITA)

should be equal to the time for the swirl to make one complete rotation, and thus the flame front is kept practically stationary, the fresh mixture approaching it with the velocity of propagation of the flame and the end mixture is formed immediately in front of the stationary flame front. A 10:1 compression ratio engine has consumed fuels with octane number ranging from 100 to zero, without knocking combustion. For part load conditions, the fuel consumption shows a maximum of about 0.35 lb./h.p. hr. which can be explained only by the fact that stratification of the charge is ensured.

(b) *Limitations of direct fuel injection*—One outstanding physical phenomenon in the mode of flame propagation in cylinders into which petrol is injected is the effect of the size of the cylinder. For good atomization the fuel jet has to travel a certain distance, so that it is disintegrated sufficiently, and this calls for the possibility of deep penetration. This explains why fuel injection scored advantages and was applied in aero engine cylinders of about 6 in. diameter in the first experiment.

On the other hand, these large cylinders tend towards knocking and one trend of thought was to have a large number of small cylinders instead of a few larger ones. For this reason in particular, and also for meeting the general requirement to investigate combustion if fuel is injected into small cylinders, tests have been conducted in the Department of Internal Combustion Engineering at the Indian Institute of Science<sup>16</sup> with an extremely small engine, an air-cooled single cylinder of 250 cc. swept volume, and a bore of 2.47 in. A study of atomization of petrol was also included.

The net result was, that at high speeds, the performance was as good as with a carburetor, and not quite smooth running was encountered only under part load conditions. The problem of increasing the compression ratio was not raised in these tests.

The distribution from the carburettor to the individual cylinders is not uniform. In that case the advantage of direct injection becomes apparent as each cylinder will be supplied to its best, so that the engine as a whole would be superior.

(c) *The self-igniting otto engine*—The fact that fuel-air mixture, if sufficiently highly compressed ignites, due to adiabatic temperature increase, is used in a small two-stroke engine (Fig. 17) which has appeared<sup>17</sup> on the Continent ; it has no external means of ignition. From the point of view of flame propagation, it is interesting to note that knocking may be avoided if the dimensions are kept relatively small and if the speed is high. The exact limitations for this principle in regard to quantitative values cannot be given yet due to lack of quantitative results.

Under the working conditions of the engine, the compression ratio for normal liquid fuels is about 13 to 16 and thus is quite similar to diesel engines. For starting however, the compression ratio must be increased considerably up to 100:1 and more as the engine is then cold and fuel vaporizing possibly incomplete. In order to deal with the very considerable pressure of combustion the engine has to be built quite sturdy. The variation in compression ratio, effected during running, is by axially moving the cylinder barrel plus

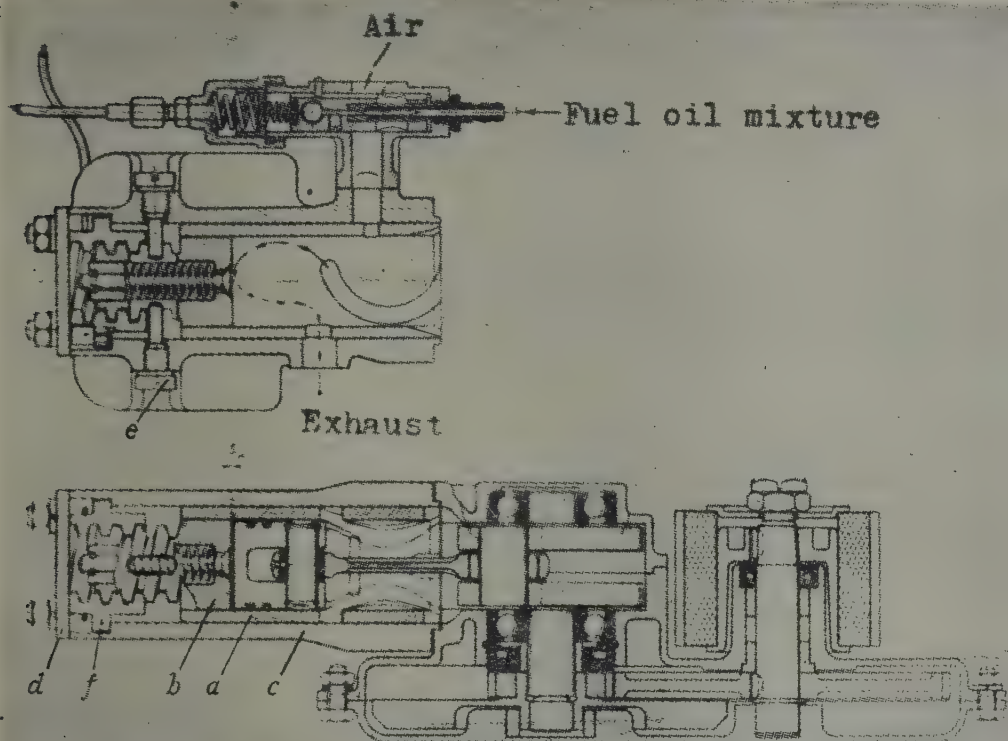


FIG. 17—SECTIONAL DRAWING OF AN ENGINE WITH SELF-IGNITION DUE TO HIGH COMPRESSION. THE ENGINE HAS A VARIABLE COMPRESSION RATIO.

the cylinder head attached to it with respect to the crankshaft. For successful running, the octane number of the fuel should be low, and under certain conditions high speed diesel oil might be used.

The speed of the engine is quite high (about 6,000 r.p.m.) and the small swept volume of 18 cc. delivers normally 0.75 h.p. output which can be brought to 1 h.p. so that the specific output is 55 h.p. litre. The best values of weight per unit output is about 5 Kg./h.p. It should, after further developments, be possible to produce such engines for smaller costs as are necessary for a new bicycle.

The principle involved in this engine might give rise to interesting further developments which in their trend will centre around the problem of balancing the dimensions of the engine and thus the output, with such means as eliminating the adverse effects of knocking combustion. The outcome of these investigations will decide whether the self-igniting petrol engine can be designed successfully for higher output and, possibly, in multi-cylinder versions. A combination with fuel injection should ultimately lead to a petrol engine of extremely high fuel economy combined with relative external simplicity and rather small expense on high duty materials and accessories.

(d) *Shock wave ignition in the pulse jet*—A particular interesting application of fundamental research on flame propagation<sup>18</sup> is the ignition of a combustible charge by a shock wave, released by the sudden explosion of

a small volume of a primary fuel-air mixture (Fig. 18). The mode of combustion is exactly that of a detonation, since the flame front is coupled to the pressure wave. It is noteworthy, however, that for sufficiently high pressure amplitudes detonation occurs also in fuel mixtures with air which normally would not detonate.

The same effects were found to occur if the pressure wave originated in the sudden expansion (caused by the rupture of a membrane) of an inert gas held under high pressure (20 kg./cm.<sup>2</sup>) and originally normal temperature. The shock wave ignites a petrol vapour air mixture, in spite of the fact, that the originally compressed gas, due to its expansion is cooled to about 100°C.

Automatically recurring self ignition by shock waves was applied in pulse jets (Fig. 19) used for the propulsion of small aircraft, working with a frequency of 38-50/sec., with gas pressures of up to 80-100 kg./cm.<sup>2</sup> The rather complicated apparatus—essentially a free piston to create the igniting shock waves (Fig. 18) was later abolished, and replaced by an arrangement (Fig. 19) which provided ignition from a pressure wave of surprisingly small amplitude entering through the outlet orifice of the jet pipe. Since a large number of different types of fuels showed the same behaviour under conditions of running, it can be concluded that shock wave ignition presents a phenomenon different from the normal process of ignition by heat input.

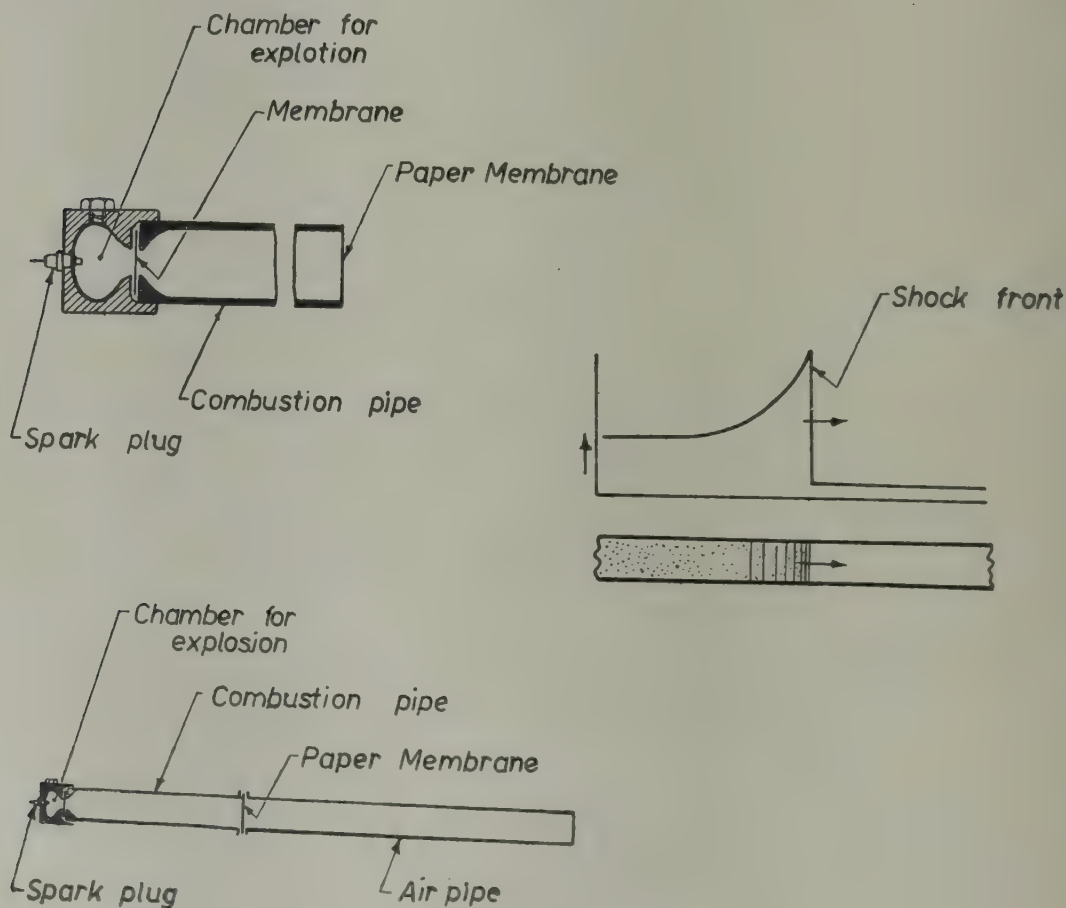


FIG. 18—IGNITION IN PETROL-AIR MIXTURES BY SHOCK WAVES

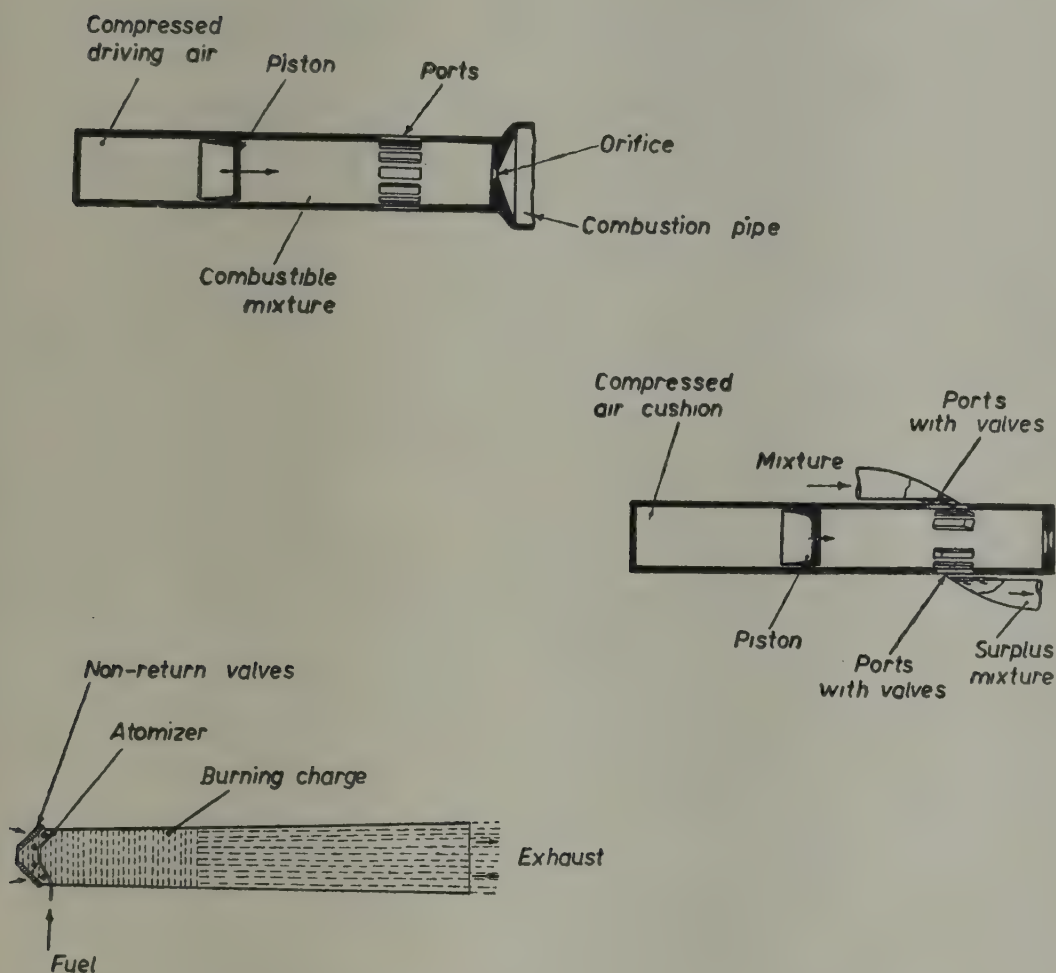


FIG. 19—CONTINUOUS IGNITION OF PETROL-AIR MIXTURES BY SHOCK WAVES

#### 4. Conclusions and Future Work

It can be concluded from the foregoing that in the field of creation of mechanical energy from the heat energy stored in fuels the mode of combustion plays the most important part. The economy of the process, for a given fuel, is largely dependent on details of the combustion process in all its aspects. A further and particularly important conclusion is given by the fact that the nature of the fuel is decisive in the process of liberation of latent energies. This is especially to be noted for fuels which are outside the scope of those investigated so far, namely, indigenous fuels, or blends of these fuels, mostly of a vegetable nature.

The blending of fundamental researches on combustion with investigations under motoric conditions will have the greatest economical and technical importance, especially in a country like India where the problem of the efficient use on a large scale of low grade fuels for power production is not yet solved.

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# Injection System for Injecting Pulverized Fuel into Pressurised Combustion Systems

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## I. Introduction

The emergence of the Gas Turbine as an economical prime mover and the continued relative high cost of oil and coal have been chiefly responsible for directing the attention of many engineers to the possible development of the solid-fuel-fired Gas Turbine power plant. It is obvious that in this country, as in many others depending on coal, the development of gas turbine prime movers will largely depend upon the ability to use solid fuels, viz., coal, in such power plants. Considering its nature and its type, the pulverised form of this fuel seems to be the best method of its utilisation in this particular field. Of the two methods of utilisation of coal, the direct method is to burn the fuel straightaway in combustors, while the indirect method is to gasify the fuel and burn the resulting gas as fuel in gas combustors. In either case, it can be seen that pulverised fuel has to be fed into systems which are operating at elevated pressures—say of the order of 2 to 5 atmos. abs. Thus there is a need to develop a suitable injection device for injecting pulverised fuel into pressurised systems.

*Present State of development*—In the United States, the Locomotive Development Committee considered both the methods of utilisation of coal for their coal burning gas turbine locomotive. In the system adopted for injecting coal to the gasifier, a special unit called the coal pressurising unit has been incorporated. This unit, works on the principle of double-cone-hopper arrangement.<sup>1</sup> For the system adopted for direct combustion of coal in locomotives, a special rotary coal pump has been introduced. Coal is fed to this pump—which is made up of a 12 in. diam. rotor with 18 small pockets—down through a feed opening at the top of the centre casing where each pocket gets filled with coal and carried around to the discharge point. It is claimed that this rotary coal pump has operated successfully.

Despite these arrangements, it can be seen that the supply of coal is intermittent. The chief requisite of an injection system should be its ability to supply a continuous and uniform mixture of coal and air to the combustion system. To achieve this, the principle of "Ejection" seems to be more plausible. This principle has been successfully applied to inject pulverised fuel into furnaces of huge steam boilers.<sup>2</sup> It should be remembered, however, that in such cases, since the pressure inside the furnace is very nearly atmospheric the application of this principle practically poses no difficulties. But the problem gets complicated once the system is expected to work under

higher furnace pressures. The present paper therefore mainly deals with the injection system—based on the principle of ejection—for a pressurised combustion equipment.

## 2. The Injection System

(a) *Principle*—The injection system under consideration incorporates the principle of 'Ejection'. In this principle, the energy of one fluid (forcing fluid) is made use of to pump a secondary fluid. This process is carried out in what are called 'ejectors'. But the term 'ejector' generally implies a type of jet pump which discharges at a pressure intermediate to 'forcing' and suction pressures. There are two types of ejectors namely (a) the venturi type and (b) the draft tube type. The draft tube type is used only for low compression ejectors. The venturi type is used for both low compression and high compression ejectors. The high compression ejector consists essentially of a 'Forcing nozzle' and a 'Diffusor'. High pressure air passing through a forcing nozzle is led into the entrance duct of the diffusor. Here, by a complicated process, secondary air is entrained and both these streams get mixed and are ejected out at the exit end. This ejection process may or may not result in a pressure rise across the diffusor. But if the diffusor body is so designed that the pressure rise is effected, a device is obtained whereby air from the outer atmosphere is accelerated into a region of high pressure. If now any particulate matter is associated with this secondary air, it should be possible to get it entrained at the high pressure end.

(b) *Design*—There are no convenient design methods known for high compression ejectors, most of the designs being based on the results of experiments. The theoretical methods of design involve the use of isentropic equations for compression and expansion of fluids.

The assumption made in the theoretical investigation is that the process is either constant-pressure mixing or constant-area mixing. In practice, however, no such rigid demarcation can be expected and the actual process is a combination of both. From a purely theoretical approach, it is found that the ejectors designed on the assumption of constant-pressure mixing process should give better performance than those designed on a constant-area basis. For a given forcing pressure ratio  $P_i/P_o$ , mass ratio  $w$ , and delivery pressure ratio  $P_3/P_o$  there exists only one value of the area ratio of the ejector which gives the best performance. These curves are based on calculations made on the assumption of constant pressure mixing—between sections X and 1 of Fig. 1—resulting in a transverse shock at section 1 and followed by subsonic diffusion between 2 and 3.

The ejectors used for experiments in the Department are designed for the following conditions:

$P_o = 14.7$  psia.  $P_i/P_o = 10$ ,  $P_3/P_o = 4$  and  $w = 0.2$ . Forcing mass flow = 1.5 lb./min. Forcing pressure is about 150 psia. The area ratio for these conditions is found to be  $a_2/a_1 = 32.5$ . The throat diameter of the forcing

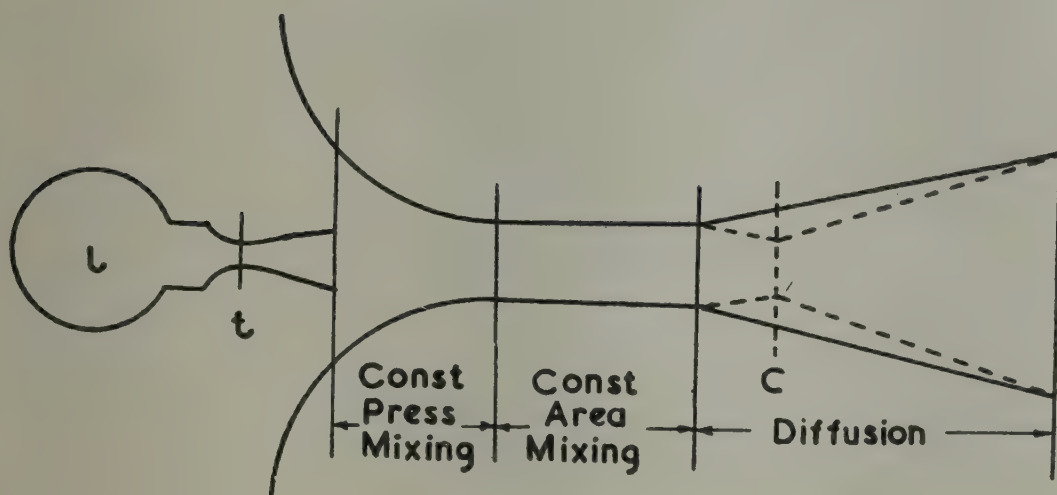


FIG. 1—EJECTOR

nozzle is 0.1 in. and that of the diffuser 0.1875 in. The entrance section of the diffuser has a converging angle of about  $4^\circ$  and the angle of divergence of the subsonic portion is of the order of  $6^\circ$ . The length of the throat of one is kept 8 times its diameter and that of the other 4 times. These are the optimum values for the design conditions.

(c) *Performance of ejectors*—The theoretical investigations merely indicate how to determine the throat diameter of the diffuser for a particular diameter of the forcing nozzle for a given set of conditions. The performance of the ejector is not only dependent upon this but also upon many other factors. It is found that a straight portion of the throat in the diffuser body gives a better performance. This is attributed to the stabilisation of the flow in this region after mixing. Here again there is an optimum length beyond which any improvement in performance due to increase in length is offset considerably by the increased surface losses. It is reported that a length corresponding to a length/diam. ratio of 6 to 8 is quite satisfactory. It is also found that the performance depends on the position of the forcing nozzle in relation to the diffuser. For each particular design there is an optimum position at which the ejector performance factors are well balanced. Figs. 2 and 3 give the general trend of the effect of distance of forcing nozzle from diffuser.<sup>4</sup> These curves show that for the particular conditions, the forcing nozzle must be withdrawn from section 1 (see Fig. 1) for good performance. They also show that there was a certain maximum and a minimum which correspond to nozzle positions regardless of the type of secondary inlet. It should be pointed out here that the performance characteristics discussed above are for ejectors working at a back pressure of nearly one atmosphere absolute. Little or no data is available on the performance of ejectors working at elevated back pressures. This is now being studied at the I.C.E. Laboratory.

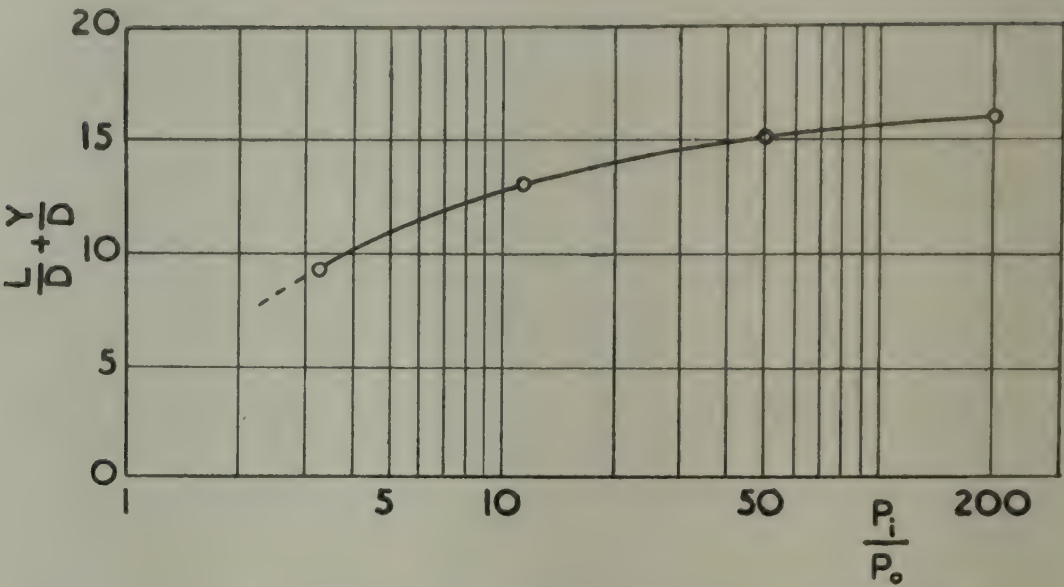


FIG. 2—BEST LENGTH FROM PRIMARY NOZZLE EXIT TO DIFFUSOR ENTRANCE EXPRESSED AS A FUNCTION OF PRESSURE RATIO

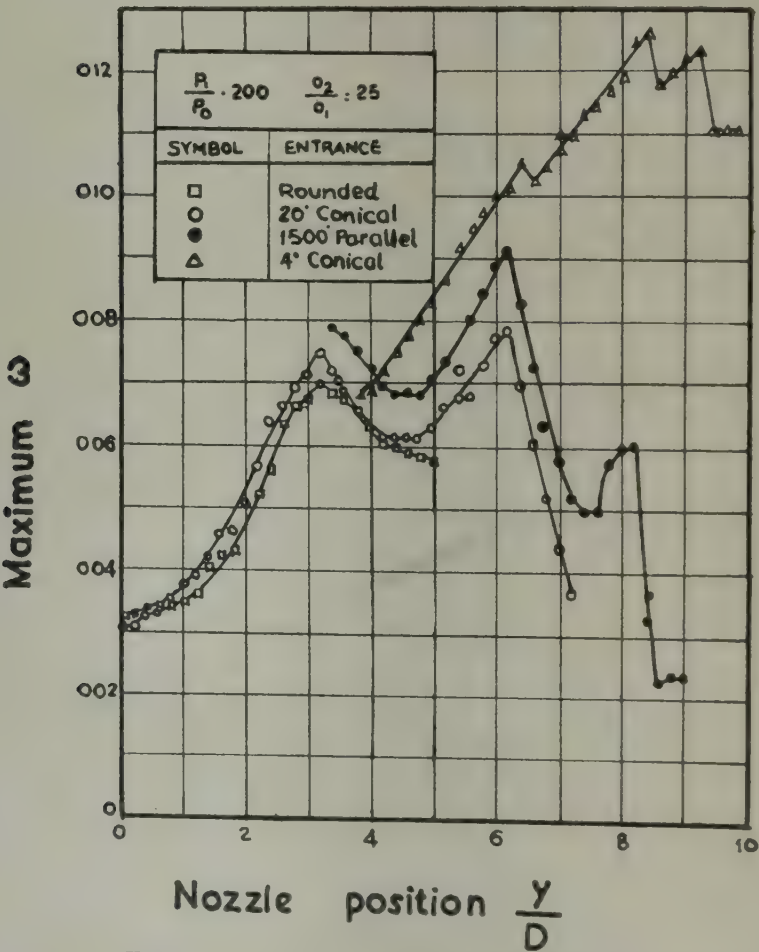


FIG. 3—VARIATION IN MAXIMUM FLOW RATIO WITH NOZZLE POSITION

### 3. Experimental Set-up

The test set-up for studying the performance of ejectors at elevated back pressures is shown in Fig. 4. Provision has been made here to vary the position of the forcing nozzle in relation to the diffuser. An orifice meter installed ahead of the forcing nozzle determines the forcing flow. Pressure tapings are provided to record the static pressures at the points X, 1, 2 and 3 indicated in Fig. 1. At the discharge end, two flow meters, one a nozzle and the other an orifice type are provided. These are meant to measure the coal and air mixture passing through. A cyclone dust collector at the exit end collects the coal dust entrained.

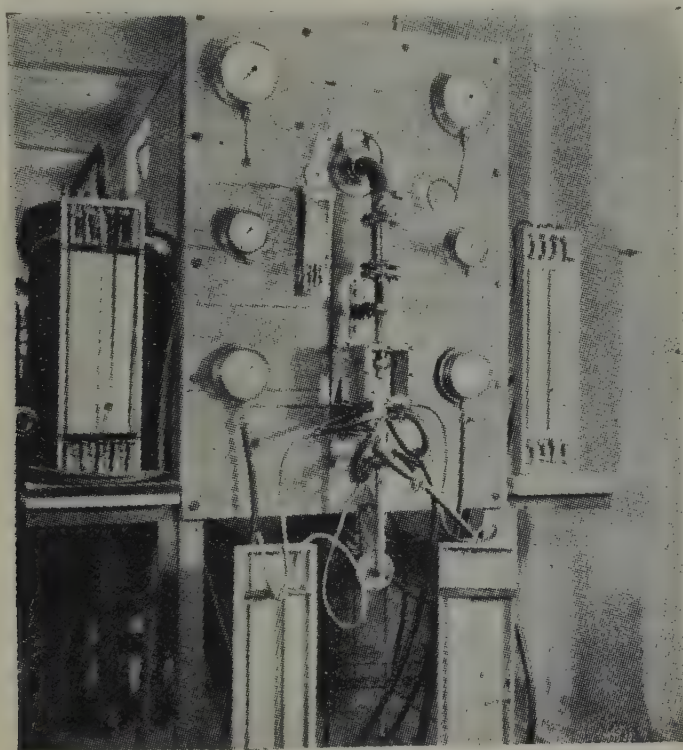


FIG. 4—TEST SET-UP FOR STUDYING THE PERFORMANCE OF EJECTORS AT ELEVATED BACK PRESSURES

### 4. Conclusion

Preliminary experiments conducted on these ejectors have indicated that a back pressure as high as 2.25 atmos. abs. could be expected with the existing design. Experiments of a detailed nature which are being carried out at present have given encouraging results. With the knowledge of the performance characteristics of the ejector discharging at elevated back pressures, it is intended to apply this for conveying coal dust. Some difficulties are likely to be met with, in the initial attempts by the shifting of shock wave due to the presence of solid particles. But it is too premature to make any general remarks regarding this at the present stage. However, what is most

necessary to be ensured is a uniform mixture of coal and air at the discharge end. In addition, the coal-air ratio should be within reasonable limits. While in the case of direct combustion a slightly marked variation in coal-air ratio may not matter much, for the gasification chamber, on the other hand, the amount of air associated with coal should be a minimum. This critical condition is likely to impose a serious restriction on this type of injection system. But in this case, the forcing fluid may be conveniently changed over to steam as this actually helps to a certain extent in the gasification process.<sup>5</sup> Since, the data on the performance of steam-air ejector discharging at higher back pressures, are very meagre, it is a secondary problem that remains to be investigated at a latter stage. The main attempt at present is to arrive at a suitable air-ejector which will deliver a uniform mixture of coal and air to supply the needs of pressurised combustion system.

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## Material Problem and Component Manufacture

The third session was held on 6 April 1952 at 2 p.m. in the auditorium of the Department of Internal Combustion Engineering. Prof. M. S. Thacker, Director, Indian Institute of Science, Bangalore, was in the chair.

Ten papers were read and discussion followed the reading of each paper.



# Spheroidal Graphite Cast Iron and its Application to Diesel Engine Parst

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Spheroidal Graphite Cast Iron, which is often referred to as Ductile Iron or Nodular Iron, is the latest and undoubtedly the most important development which has taken place in the cast iron foundry field in recent years. As the name implies, in this new type of iron the graphite is present in the form of spheroids, whereas in ordinary gray cast iron or high duty cast iron the graphite is present in the form of flakes. Typical shapes of graphite occurring in spheroidal graphite cast iron (which hereafter will be referred to as S. G. Iron) and in gray cast iron are shown in Figs. 1 and 2.

The formation of spheroidal graphite may be accomplished by the introduction of one or other of a number of different elements either singly or in combination, into molten cast iron. Undoubtedly the most effective and reliable method of producing spheroidal graphite, so far developed, is through the introduction of magnesium in the form of a suitable addition alloy. The effect on the graphite form of the introduction of magnesium into molten cast

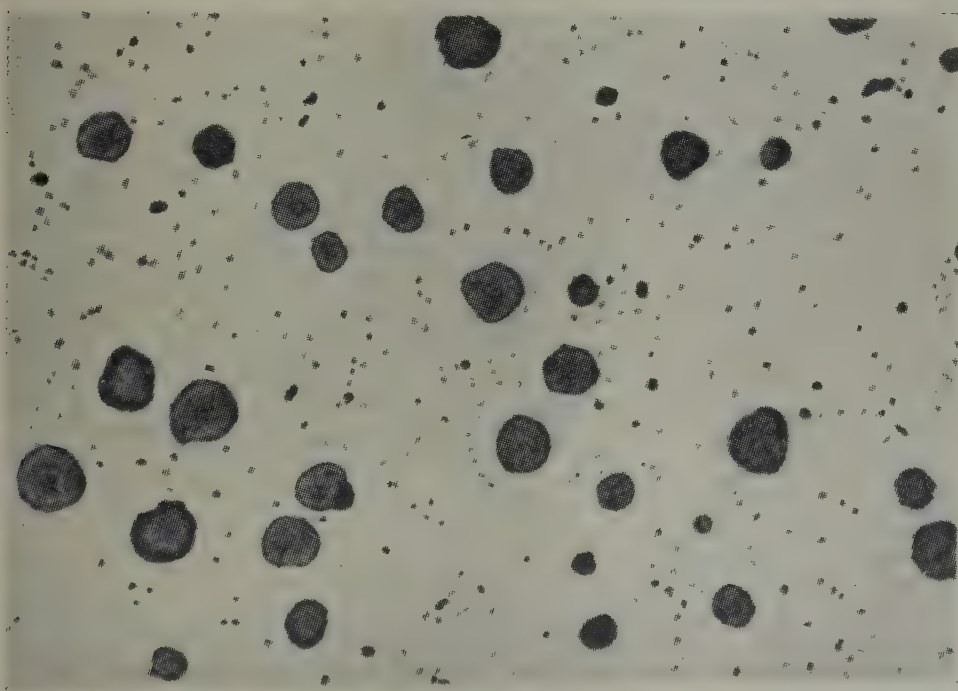


FIG. 1—TYPICAL STRUCTURE OF S.G. IRON, UNETCHED  $\times 100$



FIG. 2—TYPICAL STRUCTURE OF GRAY CAST IRON, UNETCHED  $\times 100$

iron was first discovered in the Research Laboratory of the International Nickel Co. Inc., U.S.A. The first public intimation of the magnesium process was made in May 1948<sup>1,2</sup> while the first descriptive article on the subject was published in February 1949.<sup>3</sup> Since then the development of the commercial production of castings in S. G. Iron has advanced rapidly in U.S.A., Britain and other countries. Although this iron must be considered to be in its relatively early stages the potentialities of this material have already become widely recognised and many thousands of tons of castings have been made by the 93 foundries in U.S.A. and Canada licensed to operate under patents taken out by the International Nickel Co. Inc., U.S.A., and the 71 foundries licensed to operate under patents taken out by the Mond Nickel Co., Ltd., London,\* up to the end of 1951. These patents cover the introduction of magnesium in any form to leave a residual magnesium content of a small but significant amount productive of the spheroid form of graphite as cast or after heat treatment. The range of castings so far produced vary in weight from a few ounces to about 50 tons.

#### MECHANICAL PROPERTIES

As is generally known the strength of cast iron is very largely controlled by the shape and size of the graphite. In ordinary gray cast iron and in high duty cast iron, the graphite is present in the form of flakes, these flakes having a markedly weakening effect on the iron. The size of the graphite flakes can be controlled to some extent by adjustments in composition and the rate of cooling from the molten state. The larger the flakes and sharper the ends the weaker will be the iron. By reducing the size of the flakes as is accomplished in high duty cast iron, the strength of the iron is increased. As the

\* Indian Pat. 39,302.

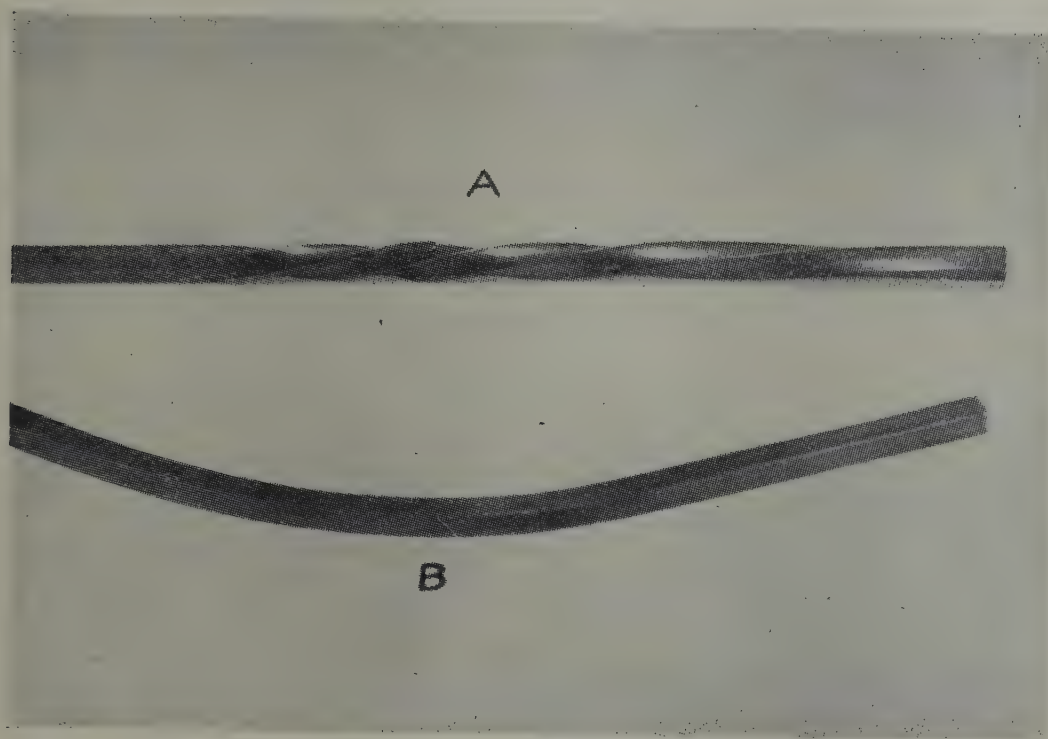


FIG. 3—(A) SAMPLE OF INDIAN MADE S.G. IRON AS-CAST, 10 IN.  $\times$   $\frac{1}{4}$  IN.  $\times$   $\frac{1}{4}$  IN.  
TWISTED UNTIL FRACTURE OCCURRED  
(B) SAMPLE OF INDIAN MADE S.G. IRON AS-CAST, 10 IN.  $\times$   $\frac{1}{4}$  IN.  $\times$   $\frac{1}{4}$  IN.  
BENT UNTIL FRACTURE OCCURRED

ends of the flakes become rounded and their size also reduced, the strength of the iron is further increased until when the spheroidal form is achieved the weakening effect of graphite is reduced to a minimum.

The strength of cast iron also varies according to the structure of the matrix and this holds good in the case of S. G. Iron. In fact the strength of S. G. Iron is largely determined by the matrix structure. In each case the tensile strength can be improved by additions of appropriate alloying elements.

One of the features which has limited the use of cast iron for engineering purposes has been lack of ductility. S. G. Iron, even in the as-cast condition, is slightly ductile (Fig. 3). After a relatively short heat treatment the ductility of S. G. Iron is materially increased (Fig. 4).

While this feature is of importance in many applications, and opens up new fields of uses for iron castings, there are many purposes for which ductility is not required but where high strength is called for sometimes combined with good wearing properties, and in this field S. G. Iron castings are finding ready application either in the as-cast condition or after a short low temperature treatment designed only to remove casting stress and not to increase ductility.

In the as-cast condition without the introduction of alloying elements, other than the amount of nickel resulting from the use of a nickel-magnetism addition alloy as the medium of adding magnesium to the molten iron, the ultimate tensile strength is of the order of 35-45 tons/sq. in. with elongation values in the range of 1-6 per cent. The corresponding Brinell Hardness numbers will be about 230-280. If the composition of the basis iron is such

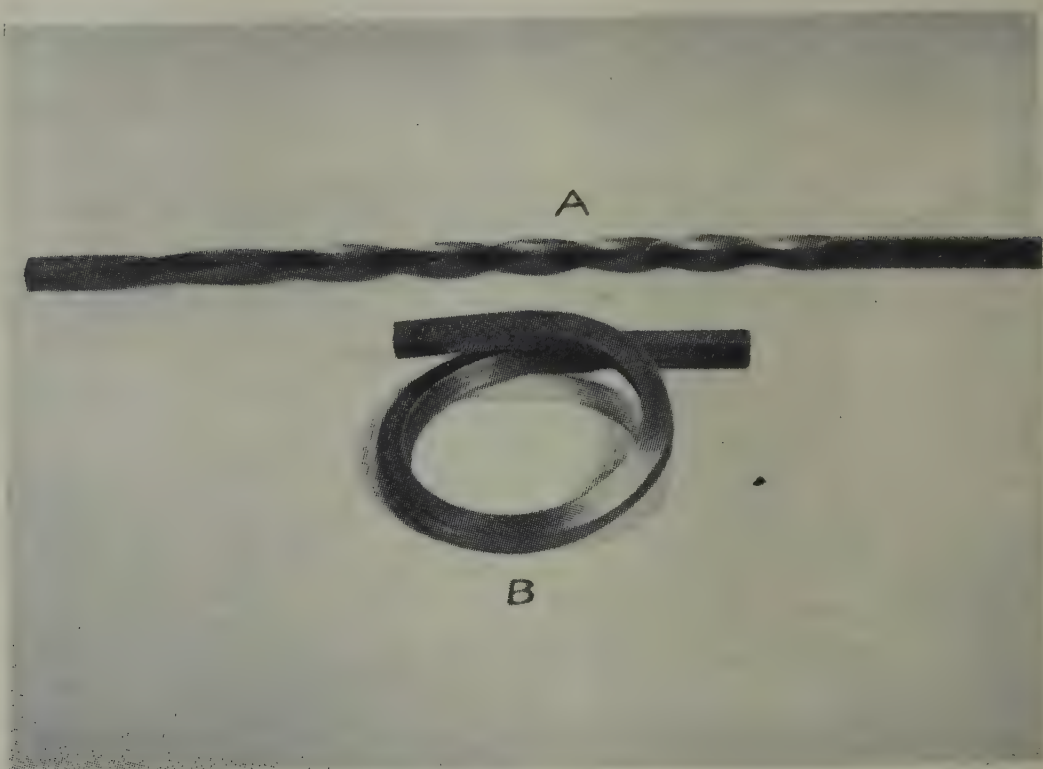


FIG. 4—(A) SAMPLE OF INDIAN MADE S.G. IRON, HEAT-TREATED, 10 IN.  $\times$   $\frac{1}{4}$  IN.  $\times$   $\frac{1}{4}$  IN.  
TWISTED BUT NOT BROKEN  
(B) SAMPLE OF INDIAN MADE S.G. IRON, HEAT-TREATED, 10 IN.  $\times$   $\frac{1}{4}$  IN.  $\times$   $\frac{1}{4}$  IN.  
BENT BUT NOT BROKEN

as to give a pearlitic matrix (as shown in Fig. 5), then the strength and hardness values will be at the upper end of the range, and elongation at the lower end.

By means of a simple heat treatment, the object of which is to convert the matrix structure to the ferritic condition (Fig. 6), the ultimate tensile strength drops to about 27-35 tons/sq. in. the elongation increases to some 10-25 per cent and the hardness drops to 140-180.

As is well known, ordinary flake graphite iron does not show a true yield point when tested in tension, but this is not so in the case of S. G. Iron which shows a distinct yield point. In the pearlitic condition the yield point comes within the range of 25-35 tons/sq. in. and 20-25 tons/sq. in. in the ferritic condition. This material retains linear proportionality of stress to strain up to high loads and is truly elastic. Its modulus of elasticity has been reported by Gagnebin, Mills and Pilling<sup>4</sup> to be about 25 million lb./sq. in. to be uniform from heat to heat, and little affected by composition and section thickness. This is of importance in engineering applications where stiffness is a desirable feature. The equivalent figure for ordinary grades of flake graphite iron ranges from 15-18 million lb./sq. in. and for steels about 20 million lb./sq. in.

The resistance of S. G. Iron to shock is so markedly superior to that of flake graphite iron that the standard form of impact test piece as used for

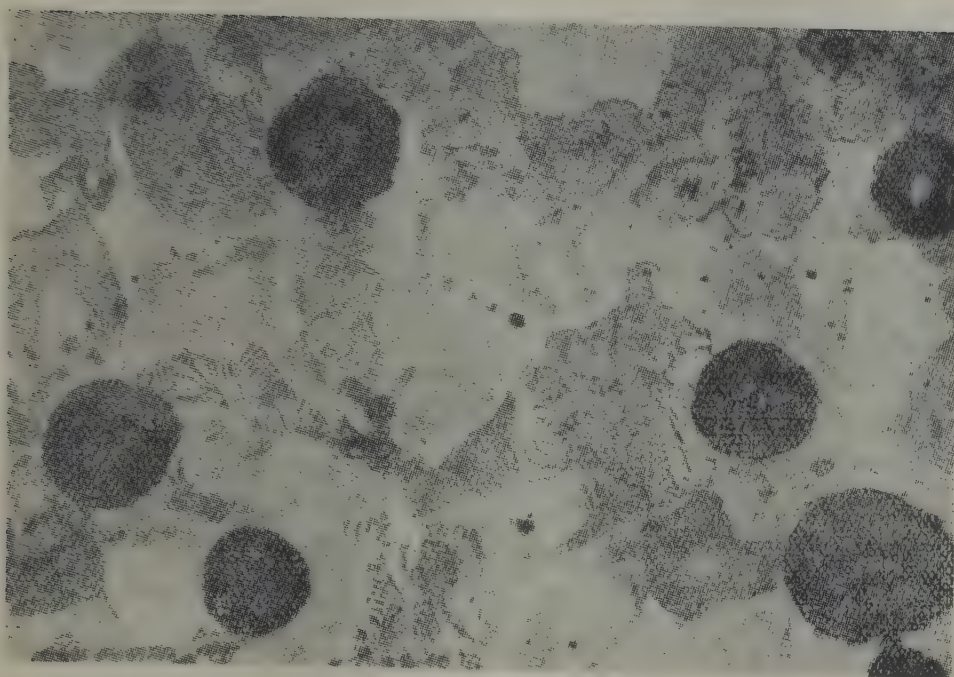


FIG. 5—S.G. IRON, AS-CAST, SHOWING PEARLITIC MATRIX. ETCHED  $\times 250$

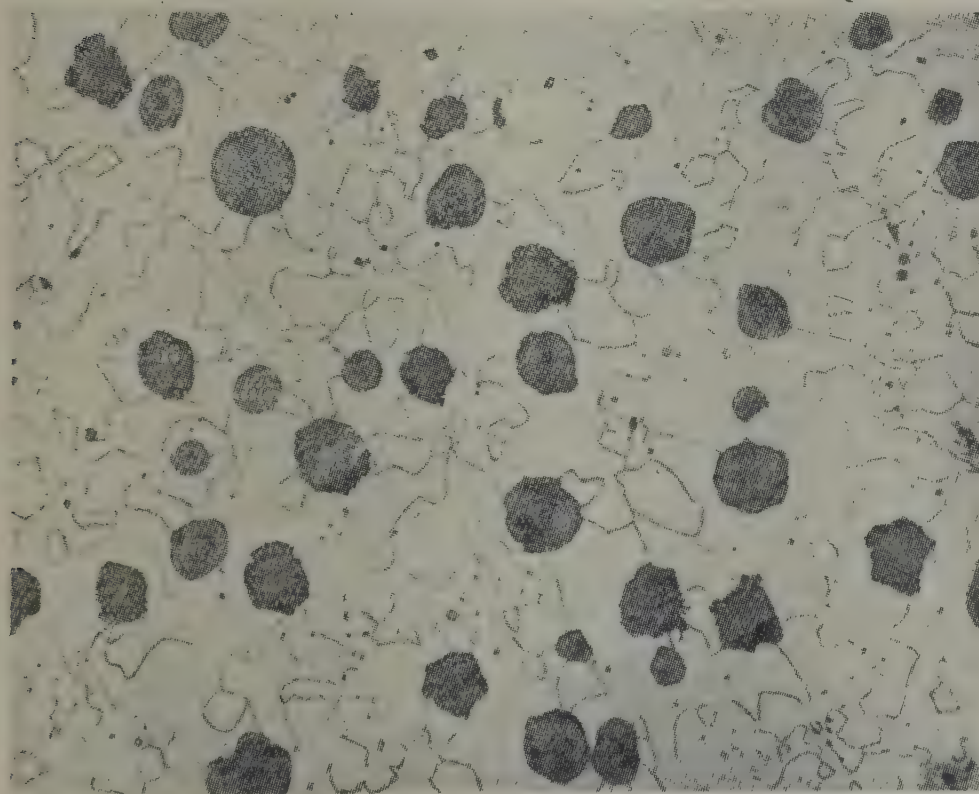


FIG. 6—S.G. IRON, HEAT-TREATED, SHOWING FERRITIC MATRIX. ETCHED  $\times 100$

cast iron (B.S. 1349), namely, an unnotched 0.798 in. diam. test piece, is not suitable since specimens with elongation values of 5 per cent or more, usually remain unbroken. When using a notched test piece 0.450 in. diam. the Izod impact value as-cast is about 3-5 ft. lb. and after heat treatment 10-15 ft. lb.

As regards endurance limit, available evidence shows that the endurance limit of S. G. Iron is substantially higher than anything previously available from flake graphite iron, and compares very favourably with many steels especially under notched conditions. As an indication of the endurance limit of S. G. Iron, figures given by Gagnebin, Mills and Pilling<sup>3</sup> in their original paper may be quoted, namely, 18 tons sq. in. in the as-cast condition and 13-16 tons sq. in. after heat treatment.

The mechanical properties of S. G. Iron and high duty flake graphite cast iron are given in Table 1.

**TABLE 1—PROPERTIES OF S.G. IRON AS-CAST AND ANNEALED COMPARED WITH HIGH DUTY FLAKE GRAPHITE CAST IRON**

	High Duty Flake Graphite Cast Iron	S.G. Iron	
		As-cast	Annealed
Ultimate tensile strength, tons/sq. in.	18-22	35-45	72-35
Yield Point, tons/sq. in. ...	—	25-35	20-25
Elongation, per cent ...	Nil	1-5	10-25
Transverse Rupture Stress, tons/sq. in. ...	38-42	55-65	55-60
Compressive strength, tons/sq. in. ...	60-65	65-80	—
Comparative yield strength, tons/sq. in. ...	—	32-40	24-32
Elastic Modulus, lb./sq. in. ...	$18 \times 10^6$	$25 \times 10^6$	$25 \times 10^6$
Brinell Hardness ...	210-240	230-280	140-180
Impact Izod 10 mm. sq. notched, ft. lb. ...	1	4	12
Endurance Limit Unnotched, tons/ sq. in. ...	8.5	13-18	11-13
Endurance Limit Notched 0.05 in. rad., tons/sq. in. ...	8.0	9.5	8.2

### Specification

From what has already been said it will be evident that the term S. G. Iron does not signify cast iron of any one specific combination of properties but rather a new type of iron having mechanical properties superior to those previously obtained and covering a considerable range exceeding certain minima. While standard specifications have not yet been issued, in order that engineers, designers and others interested in the adoption of S. G. Iron for specific purposes may be guided, tentative specifications have been drawn up by the Mond Nickel Co. Ltd., covering three grades. Details of these three grades are given in Table 2. That these grades can be made in India, is now established.

**TABLE 2—TENTATIVE SPECIFICATIONS FOR S.G. IRON**

	Pearlitic	Pearlitic/ Ferritic	Ferritic
Ultimate tensile strength, tons/sq. in. ...	37 min.	32 min.	27 min.
Yield Point, tons/sq. in. ...	27 "	24 "	20 "
Elongation (%) ...	1 "	5 "	10 "

All as determined on 1 in. thick keel block test bars, cooled in the mould and/or annealed.

### **Machinability**

S. G. Iron both in the as-cast (pearlitic) condition and in the heat-treated (ferritic) condition is readily machinable and takes a very high finish more resembling that of steel, as would be expected from the graphite structure of the iron. Taking into account the fact that S. G. Iron is harder than flake-graphite iron of the same matrix structure, it might be assumed that it would be slightly harder to machine. That this is not so has been demonstrated by investigations carried out in U.S.A.<sup>5</sup> and confirmed by engineering firms who have machined numerous castings in both of the above stated conditions. As-cast S. G. Iron having a tensile strength of 41.5 tons per square inch was found to have the same machinability rating as flake graphite iron with a strength of 20 tons per square inch. In the heat treated condition this new iron can be machined at a rate 2 to 3 times that of good quality flake graphite iron.

### **Wear Resistance**

In order adequately to assess the wearing properties of any material lengthy service trials are necessary. As S. G. Iron is a relatively new material some time will have to elapse before its wear resisting properties can be placed in proper position in relation to other materials. However, service results to date indicate that S. G. Iron shows a notable resistance to wear under lubricated conditions.

Gagnebin, Mills and Pilling<sup>4</sup> have reported results of gall tests carried out in the Research Laboratory of the International Nickel Co. Inc. This test consisted of rubbing a cylinder against an annular ring of the same material without lubrication and under conditions of increasing load. Their results showed that S. G. Iron is not inferior to flake graphite iron in resistance to galling.

Wherever resistance to wear is a desirable feature then the matrix structure of the iron should be substantially free from ferrite.

### **Resistance to Heat**

Under exposure to elevated temperatures ordinary flake-graphite iron is subject to growth and oxidation which is influenced by the shape of the graphite flakes. These flakes which are often interconnecting provide a means of penetration of air or other gases resulting in oxidation and growth. When graphite is present in the spheroidal form the degree of interconnection between the graphite spheroids is reduced to a minimum, thereby limiting the ease of penetration of air or gases. This fact alone would suggest that the resistance of S. G. Iron to deterioration when exposed to heat would be superior to that of flake-graphite iron, and this has been borne out in service trials and laboratory tests. Further, Eagan<sup>6</sup> has shown that under the conditions of laboratory tests reported by him, involving cyclic heating to 1,650°F. (898°C.) and cooling, there was material deterioration in the mechanical properties of flake graphite iron whereas the tensile strength of S. G. Iron was actually increased though the ductility was lowered.

## APPLICATIONS TO DIESEL ENGINE PARTS

The foregoing review of the properties of S. G. Iron, though not exhaustive, will probably serve as a basis on which to assess the suitability of this iron for some specific applications in the diesel engine manufacturing field. It is possible that S. G. Iron may be used in place of steel for some items. Advantage might also be taken of the increased strength of this material to reduce section thickness of castings while its wearing properties and improved resistance to heat are of interest for some applications. Although detailed service results are not yet available, reports to date on several applications referred to hereunder are very favourable.

*Crankshaft*—This is probably the application of major interest to diesel engine manufacturers at the present time. The use of a cast crankshaft in oil, petrol and gas engines, and compressors, dates back many years, but the more general trend towards the adoption of cast crankshafts followed the development of the higher strength special cast irons. Many papers have been published dealing with this subject and those interested are referred to a very excellent review of the literature up to the end of 1947 prepared by Love.<sup>7</sup> Everest<sup>8</sup> in reviewing the use of cast iron crankshafts, in May 1951, stated that in Britain, in the diesel engine field, cast iron crankshafts had been used in sizes up to 10 in. diam. journals and were on trial in sizes up to 12 in. to 14 in. diam. journals, also that diesel engines were in regular production, with cast iron crankshafts, in sizes up to 2,400 h.p. and more. They had been widely adopted for diesels in stationary plant, marine service and in some agricultural applications including tractors. In U. S. A. very wide use has been made of cast iron crankshafts even to a greater extent than in Britain. Finlayson,<sup>9</sup> in describing production of cast iron crankshafts by Pacific Car & Foundry Co., U. S. A., in 1950, referred to shafts weighing 500-5,000 lb. and in overall lengths from 6-18 ft. which is indicative of the state of development of cast shafts in that country.

Just prior to the development of S. G. Iron attention was being focused on nickel-molybdenum cast irons, generally known as 'Asicular cast iron'.<sup>10</sup> Following the successful production of S. G. Iron castings, on a commercial basis, attention has been directed to the use of this material for cast crankshafts, and judging from the growing interest which is being shown the indications are that this development is proceeding quite satisfactorily and that good results are being obtained. In fact S. G. Iron is now actually specified for crankshafts, for some designs of diesel engine. Most of the S. G. Iron shafts which have been put into service have been used in the as-cast condition or in some instances after a stress relief anneal.

In the review by Everest,<sup>8</sup> previously referred to, he has given a comparison of the properties of some typical crankshaft materials. The data contained in Table 3 was taken from this source. With reference to the figures quoted, as pointed out by Everest, it is extremely difficult to give strictly comparable figures, but those included in the table will serve to convey an

appreciation of the suitability of this new type iron for the application in question, from the aspect of mechanical properties.

TABLE 3—SOME TYPICAL PROPERTIES OF CRANKSHAFT MATERIALS

	Forged Carbon Steel quenched and tempered	Ni.Cr.Mo. Alloy Steel, forged, heat- treated	High Duty Cast Iron to 20 ton 'Tensile minimum	Acicular Cast Iron	Spheroid- al Gra- phite Cast Iron (Pearlitic)
Tensile strength, tons/sq. in.	36	58	22	26	40
Yield strength, tons/sq. in.	20	50	—	—	30
Elongation, per cent	25	23	1	1	2
Brinell Hardness	160	269	220	280	260
Modulus of Elasticity, lb./ sq. in. $\times 10^{-6}$	29	29.5	18-20	22	25
Modulus of Rigidity, lb./ sq. in. $\times 10^{-6}$	11	11.65	8.6	8.6	—
Izod Notched Impact value, ft. lb.	30	60	1	1.5	3.5
Endurance Limit Unnotched, tons/sq. in.	15.8	29	7.2	12	15*
Endurance Limit 0.05 in. radius groove, tons/sq. in.	8	17.5 <sup>1</sup>	6.0	9	10*
Reduction of Endurance Limit due to groove, per cent	49	40 <sup>1</sup>	16.7	25	33*
Vibration damping capacity <sup>1</sup> 60° Sharp Notch	Low	Low	High	High	Medium

\* Amended figures

Love<sup>7</sup> has stated that the more important mechanical properties of materials for crankshafts are fatigue strength, notch sensitivity, modulus of elasticity and modulus of rigidity. As regards fatigue strength, S. G. Iron is superior to acicular cast iron, which has been proved satisfactory in service, but its endurance limit is less than that of alloy steel. On the other hand available data indicates that the notch sensitivity of S. G. Iron is less than that of steel. The modulus of elasticity of S. G. Iron is also higher than that of acicular cast iron although lower than that of steel. Unfortunately no data is available yet regarding modulus of rigidity.

It has already been indicated that the wear resistance of S. G. Iron is not inferior to that of flake graphite iron and as experience over many years, with flake graphite cast iron crankshafts has proved that the bearing properties of those shafts are excellent and generally superior to those of steel shafts, S. G. Iron should be perfectly satisfactory from this aspect. Service reports go to confirm that such is the case.

Probably the main reason why cast iron crankshafts have come into wide use is the fact that in most cases they can be produced more cheaply and more quickly than steel crankshafts. The cost of patterns is less than the cost of forging dies, and production quicker. Cast iron castings can be produced to closed tolerance to size thereby reducing the weight of metal to be machined off with consequent saving in machining time and metal. Further, in some cases the weight of the finished shaft can be reduced by coring out of journals and bearings. As crankshafts can be cast accurately to shape, this leads to

additional scope in design. One typical instance of economy of material and reduction in machining time quoted among others, by Love<sup>7</sup> might be repeated here. It relates to the Ford V-8 crankshaft which in the form of a rough casting weighed 65 lb. and the finished shaft weighed 56 lb. When made from a steel forging the rough forging weighed 82 lb. and when finished machined 66 lb. The number of production operations was reduced from 62 to 54 when the change was made to the cast crankshaft.

As illustrative of this application finished machined crankshafts cast in S. G. Iron are shown in Figs. 7 and 8. The one illustrated in Fig. 7 was the fore-runner of several thousands already in service in 6 h.p. diesel engines.

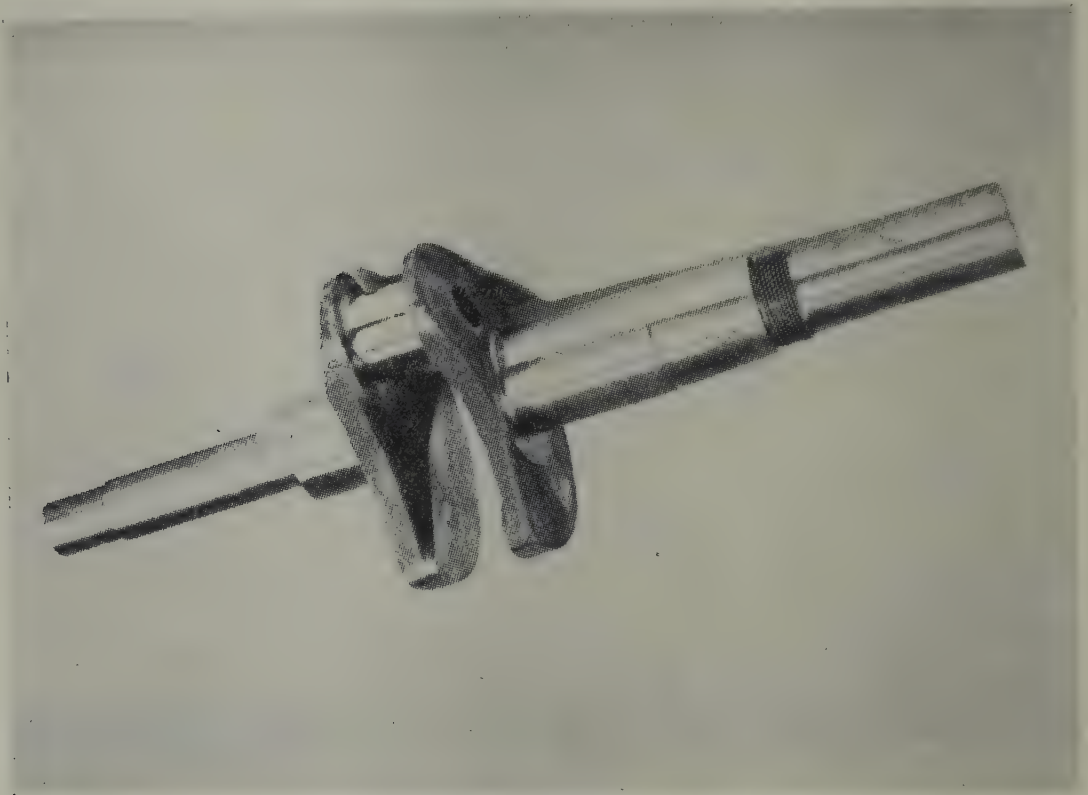


FIG. 7—SINGLE THROW CRANKSHAFT, CAST IN S.G. IRON  
(Courtesy: Sheepbridge Engineering Ltd., England)

*Cylinder Heads*—Casting of cylinder heads is a very promising use of S. G. Iron. Reference has been made to the superior heat resisting characteristics of this iron, a feature which indicates its advantageous adoption for cylinder heads. It has recently been reported from U.S.A. that heads which were giving trouble with heat craking are lasting at least five times as long when made in S. G. Iron. On the Continent, interest is being shown in this new iron for cylinder heads for heavy duty truck type diesels. A typical head in this class, cast in S. G. Iron, is shown in Fig. 9.

*Cylinder Liners*—In view of the experience which is accumulating in support of the statement made already that under lubricated conditions the wearing properties of S. G. Iron are good, it would be natural to anticipate that this material would be well suited for use for cylinder liners. Extended

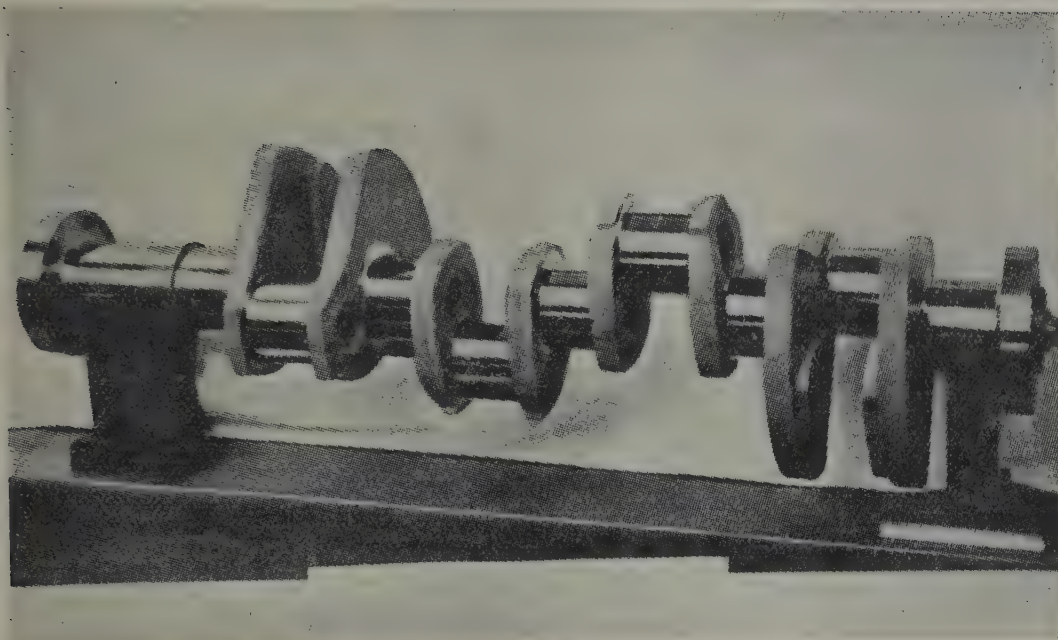


FIG. 8—FOUR-THROW DIESEL ENGINE CRANKSHAFT, CAST IN S.G. IRON  
(Courtesy: The Cooper Bessemer Corporation, U.S.A.)

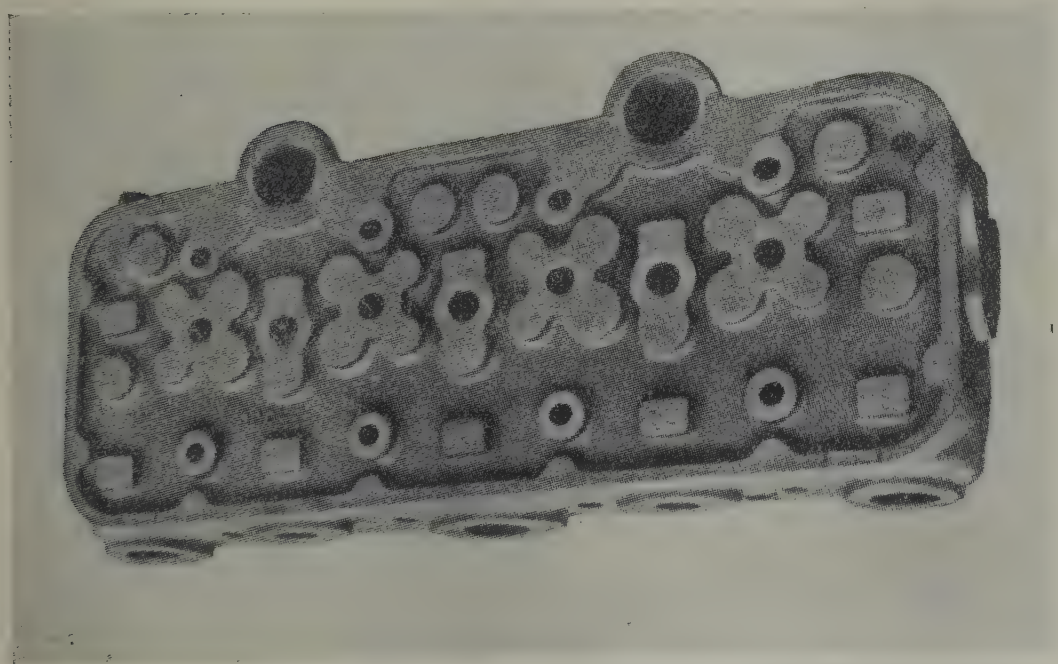


FIG. 9—DIESEL ENGINE CYLINDER HEAD, CAST IN S.G. IRON  
(Courtesy: Fonderia Guido Glissentì Fu Francesco, Italy)

tests are now being carried out with S. G. Iron liners and also piston rings, but so far no definite results are available. It is possible that due to the increased strength obtainable in S. G. Iron, at least in some instances, a reduction in wall thickness may be made thereby securing more efficient cooling.

*Pistons*—In U. S. A. this new iron has been made use of for pistons and it is under consideration for this purpose in Britain. The high strength and heat resistance of S. G. Iron would point to its suitability for this part

*Rocker Arms*—While these parts are normally made in the form of steel stampings, they have often been made in malleable iron and considerable success has been achieved in the use of acicular iron castings. As the properties obtainable in S. G. Iron are superior to those of malleable iron or acicular iron its use for this purpose is worthy of consideration.

*Exhaust Manifold*—This is another instance where heat resistance is of some importance. Successful test results with the use of this new material for exhaust manifolds has been reported from U.S.A.

*Timing Gears*—While no specific data is available relating to the use of S. G. Iron for timing gears and other auxiliary gearing on diesel engines, there is no doubt that it could be applied successfully for these parts. Apart from diesel engine practice, S. G. Iron has proved very successful for a wide variety of gears, some typical instances of which are shown in Fig. 10. Sheley<sup>11</sup> has quoted the successful replacement of carbon and alloy steel castings and forgings by S. G. Iron for gears and reports its wear resistance to be excellent. At Black-Clawson Co., Hamilton, Ohio, U.S.A., S. G. Iron gear castings are in production ranging in weight from 15-1,500 lb.

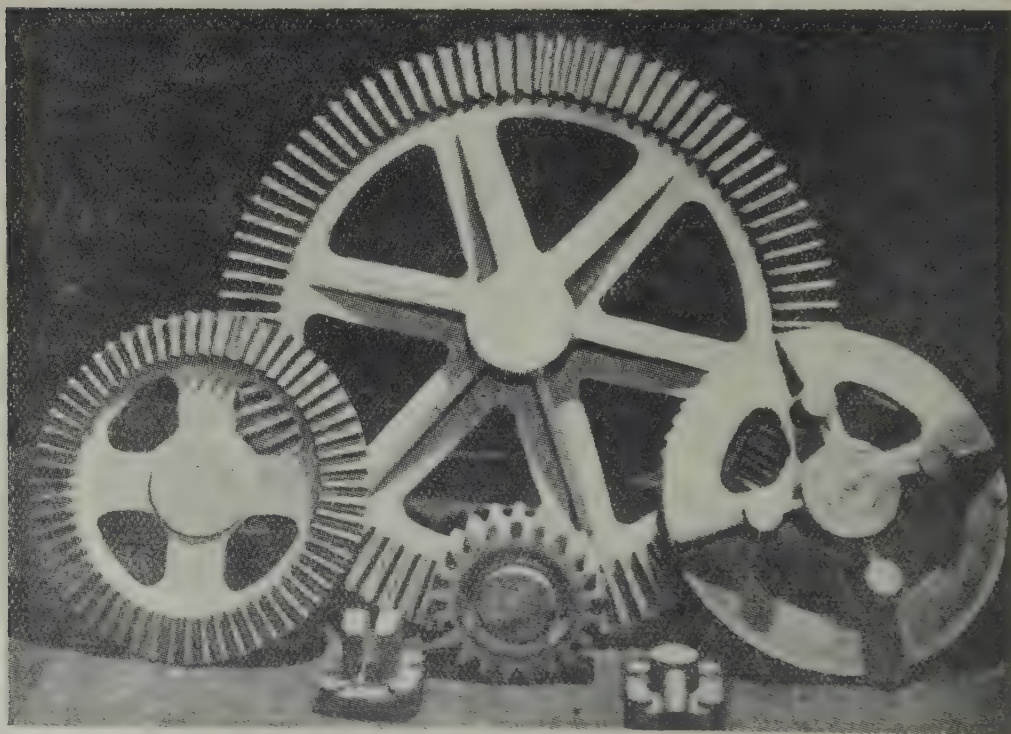


FIG. 10—GEAR AND SPROCKET WHEELS, CAST IN S.G. IRON

(Courtesy: Robert Taylor (Ironfounders) Ltd., Scotland)

*Fuel Injector Bodies*—This is an application for which the use of S. G. Iron has been studied extensively in Britain, with a view to replacing parts machined from solid steel. Although this development has not yet reached the stage of ultimate success there is every reason to anticipate that such will be the case.

*Miscellaneous Items*—Among the other items for which S. G. Iron would appear to be suitable in connection with diesel engine construction, there could be listed pulleys, miscellaneous brackets, bearing shells, starter housings, clutch parts, etc. Fig. 11 shows a flexible coupling cast in S. G. Iron. While this coupling is larger than would be used for the range of diesel engines now being made in India, the illustration is included as indicative of this application.

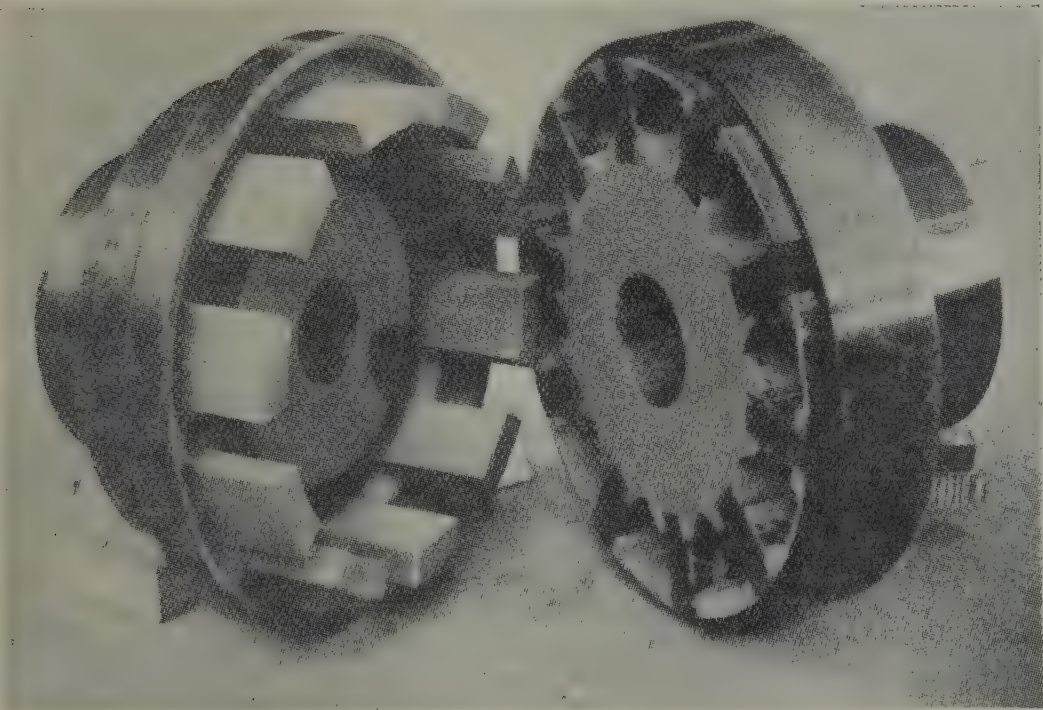


FIG. 11—FLEXIBLE COUPLING CAST IN S.G. IRON

(Courtesy : Fonderia G. Tagliabue, Italy)

So far reference has been made only to S. G. Iron in the as-cast or annealed condition as these are the conditions in which the bulk of castings are in use. By oil-quenching and tempering the range of mechanical properties obtainable can be extended, while by flame hardening the wearing properties can be greatly improved. Fig. 12 shows a section of a sprocket wheel cast in S. G. iron with flame-hardened teeth. Further, by casting against a chill a combination of a hard surface with ductile backing can be secured in S. G. Iron, a feature not procurable in any other material.

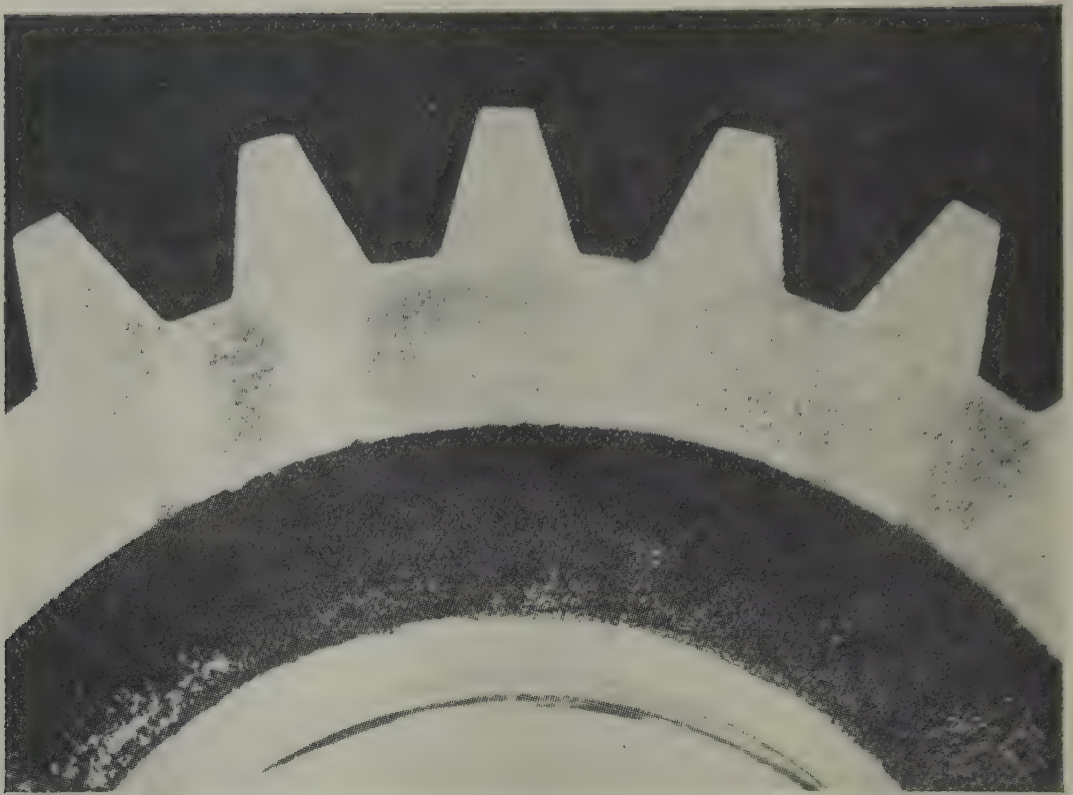


FIG. 12—CHAIN SPROCKET CAST IN S.G. IRON WITH FLAME-HARDENED TEETH

### CONCLUSION

The full utilisation of this new iron in diesel engine construction is still far from being achieved, and so the subject could only be dealt with in somewhat general terms. At the same time it is thought that given a broad outline of the properties of S. G. Iron and its potential in this field, diesel engine manufacturers in India might gain some guidance as to where advantage could be taken of the properties obtainable in this material which is helping to fill the gap between cast iron and steel, and in some cases, successfully replacing steel parts.

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# Meehanite High Duty Irons and Their use in The Automobile Industry

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Research and development in metallurgical science has evolved a great variety of alloy steels, each having different properties, rendering one class of steel more suitable than another for a particular duty. Similarly, different processes of heat treatment are necessary for different steels, giving desired properties for satisfactory behaviour under particular service stresses.

Just as we have these different varieties of steel, so also we have different qualities of iron castings, each suited for a specific duty.

Although the major portion of structural material built into the modern automobile is of one kind of steel or another, iron castings play a very important role particularly with respect to the power unit.

In the earlier days of internal combustion engineering, iron castings were more or less considered to be of one quality only, which was generally referred to as gray iron. No doubt the chemical analysis and physical properties of these earlier iron castings varied from one producer to another, but in those days little importance or attention was given to this. Today the situation is entirely different.

With very few exceptions cast iron is the universal material for automobile engine cylinder blocks and cylinder heads, these being the two principal iron castings.

Some of the service conditions these two important castings have to fulfil are: (1) the components should cast readily; (2) the castings should be homogeneous and free from blow holes; (3) they should permit of rapid and accurate machining; (4) the castings should be as hard as practicable taking into account the factors mentioned above; (5) the slightest porosity must be avoided; (6) they should give maximum life from the bores of the cylinder liners; (7) the castings must be pressure tight; (8) they must retain their shape under variable thermal and mechanical impact conditions and multifarious stress application; (9) the metal must be of high tensile strength thereby allowing minimum weight with good factor of safety; and (10) The casting must above all be very rigid to successfully withstand operating conditions of the complete engine.

The above is rather a formidable list and not by any means complete, but sufficient to illustrate the importance of casting quality demanded.

Ordinary grey iron castings will no longer satisfactorily meet these requirements and therefore, a number of special High Duty cast irons have come into prominence, having chromium, molybdenum, vanadium and copper in

addition to carbon, silicon, manganese, sulphur and phosphorus, whereby the irons are given special properties to meet particular conditions relating to corrosion, heat, erosion or wear, and superior general engineering duty.

One of the most popular types of Heavy Duty irons is known under the Trade name 'Meehanite'. This metal has for many years been manufactured in India by M/s. Cooper Engineering Ltd., Satara Road, who are the sole licensing agents for the manufacturing process in India, Pakistan, Burma and Ceylon.

Coopers have also issued licenses for manufacture to the well known concerns, Messrs. Jessop & Co., Ltd., Calcutta, Binny's Engineering Works Ltd., Madras, and Premier Automobiles Ltd., Meehanite Foundry, Wadala, Bombay.

Meehanite metal is a special cast iron produced in foundries under manufacturing rights granted by the Meehanite Metal Corporation, New-Rochelle, N.Y., and the International Meehanite Metal Co., Ltd., London. It represents a discovery in the application of various metallurgical principles in the production of cast iron, these principles being applied throughout the entire manufacturing process and involving accurate regulation of type and quantity of raw materials, of melting process, of pattern making and moulding—in fact, each and every step of the manufacturing process.

Because the manufacture of Meehanite castings is based on strict control of the actual physical make-up of the iron instead of a more or less indefinite attempt to control chemical analysis, the actual metallurgical structure of Meehanite castings is predetermined in molten metal, and the metal structure itself is controlled and regulated so as to produce the exact physical properties required of the castings by the job they are called on to perform. Carefully selected raw materials are charged into a special Meehanite invention—the Equiblast Cupola—and then each ladle of the molten iron is tested so that its structural composition is known and may be adjusted before the iron is poured into the mould. Thus, the entire manufacturing procedure is scientific and, as a result, the castings provide greater assurance of dependability, uniformity and extremely close adherence to the engineering specifications.

The primary constituents of steel usually consist of pearlite and ferrite, with the exception of the hard carbon tool steels which frequently are made up wholly of pearlite. On the other hand, cast iron, malleable iron, and Meehanite Metal all contain graphite in varying forms and quantities. Cast iron is usually made up of graphite, pearlite, ferrite and iron phosphide, while malleable iron is made up of graphite and ferrite. The arrangement and quantity of these materials are the chief contributors to the physical and engineering properties of the finished product.

In Meehanite castings, both the arrangement and quantity of the pearlite and graphite are controlled during production to provide the required physical properties. It is by this means that it is possible in the general engineering types, for example, to have as-cast tensile strengths ranging up to 55,000 lb./sq. in. with other corresponding strength figures.

None of this is theory, for it has all been worked out scientifically and

proved in result, and it is this assurance of dependability and security provided by the knowledge that definite specifications can and will be met, that makes Meehanite castings unique in their field. It is this ability to control that is the source of the Meehanite slogan "Meehanite means better castings".

Perhaps the chief advantage the users of Meehanite castings obtain is the uniformity of their castings and the unusually high freedom from defects which necessitate rejections. Because of the control in manufacture, Meehanite castings, whether produced in small or large quantities, will be found to be uniformly machinable and this same control reduces to an absolute minimum casting flaws, blow-holes, hard spots, and other factors which might result in wasted machining time or production interruptions. In addition to this, the uniform structure of the metal frequently permits substantial increases in machining speeds, with resultant savings in time and cost which are always important. It is generally true also, that Meehanite castings require less finish metal, and machine at high speeds and deep cuts with sintered carbide tools, or machine smoothly and rapidly with tools of ordinary tool steel.

The ability to heat-treat and flame-harden Meehanite castings and uniformly better strength, toughness, and hardness are of importance from both the design and engineering standpoints.

With regard to the fundamentals of machine design and engineering construction, the range of properties available in Meehanite castings is most important. Meehanite castings have been described as "Bridging the gap" between cast iron and steel because they offer industry a metal which actually combines the better properties of both steel and cast iron.

Meehanite castings definitely offer all these properties in combination. Furthermore, their engineering characteristics are stable factors, permitting accurate and trustworthy design.

#### RECOMMENDED GRADES OF MEEHANITE HIGH DUTY IRON CASTINGS FOR VARIOUS SERVICE REQUIREMENTS

Twenty one standard grades are available to Industry. Seven grades suitable for General Engineering castings, having tensile strength ranging from 30,000 lb. up to heat treated castings giving over 70,000 lb./sq. in. and Brinell hardness from 193 to 600.

In wear resisting grades, 4 types are available with tensile strength ranging from 30,000 lb. up to heat treated grades of 70,000 lb./sq. in. and Brinell hardness 196 up to 600.

In heat resisting range 5 grades are available suitable for temperatures up to 1,650°F. Tensile strength range from 38,000 lb. up to 50,000 lb./sq. in. and Brinell hardness from 196 to 500.

Corrosion resisting grades are available in 5 types, each grade suited to a particular need. Tensile strength up to 45,000 lb./sq. in. and Brinell hardness from 187 upwards as required.

From the large variety of available grades it will readily be appreciated that the best possible grade of castings can be utilised to meet the specific service conditions and this is a very vital asset when considering the application of High Duty irons to the automobile industry.



# Ceramics—A Survey of their Possibilities as Gas Turbine Materials

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Secretary, I.C.E.R. Committee, C.S.I.R.

In comparison with metals, ceramics are immediately attractive because of their freedom from oxidation and more generally, their pronounced chemical stability at high temperatures. Ceramics immune to chemical attack during prolonged exposure at  $1,000^{\circ}\text{C}$ . under combustion conditions can readily be selected. Their low density (seldom more than 4) is also an attractive feature to minimise centrifugal stresses in parts rotating at high speed. On the other hand, the high mechanical strength requirements of the gas turbine parts which must be maintained (or only slightly reduced) at the operating temperature, together with virtual freedom from creep, at once restrict the materials at present available to narrow fields.

All common commercial refractories and ceramics are based on mixtures of some major and several minor components. During the burning process, needed to produce the ceramics, eutectics are formed which melt to give liquid phases in addition to the solid crystalline phases. When the material is cooled, the liquid often persists as a glass. The strength of this type of ceramic, at both normal and high temperature, is due to the inherent strength of both crystals and the glass or liquid bond. As the temperature is raised, the properties of the liquid phase soon become dominant and as a result the material changes from a state of nearly perfect elasticity to one of plasticity. For most commercial materials this transition is effective below  $1000^{\circ}\text{C}$ ., when the material begins to show pronounced creep, centred in the liquid phase, but probably much modified by the crystalline components.

While the strength of many materials of this type is adequate at room temperature, it is soon lost on heating and, in addition, creep becomes excessive. The corollary is that ceramics of suitable strength and creep resistance at high temperature will be found only in (a) multi-component materials forming eutectics of very high melting point; (b) single-component high melting point materials giving theoretically no liquid below the melting point of the component itself.

Class (a) consists mainly of the less common materials with melting points in the range  $2,000$ — $3,000^{\circ}\text{C}$ . and much more information on the properties of such mixtures is required. On the other hand, there is now a good deal of data and experience of class (b) materials, that is, the sintered oxide type of refractory.

This type of refractory is produced from single substances (usually metallic oxides) by firing the finely pulverised and pure material at temperatures

well below its melting point, when recrystallisation occurs to give a mass of tightly packed crystallites. The strength of the material is due to inter-crystalline bond forces and not to interstitial glass, and its behaviour at high temperatures depends on the inherent properties of the crystal. Aluminium oxide is the best known and perhaps the most outstanding example of the sintered oxide refractory ; compacted powders with an average particle size of about  $5\ \mu$  can be sintered at  $400^\circ\text{C}$ . below the melting point to give an intensely hard and strong product.

Drawbacks in ceramics include brittleness, i.e., the virtual absence of ductility and abrasiveness. In general they cannot be machined or built up from stock, although shaping and turbing of unfired or lightly fired material is sometimes possible with small articles. Usually the materials must be fabricated by what are, in effect, powder methods and then burned at high temperatures. Casting of the fused material is a possible alternative, but necessitates very high temperatures and introduces certain other difficulties. In the case of turbine parts, the product would need some final machining to obtain the requisite close tolerances, but with diamond tools this should present little difficulty.

The low ductility is a serious disadvantage in several respects. Such materials are liable to break down as a result of local stress concentrations—the ability of metals to yield and distribute high localised stresses being notably absent. Whilst the tensile strength of certain sintered oxides appears to be high enough to meet estimated stresses under turbine conditions, their high values are obtained only when great care is taken in testing to ensure axial loading and thus to avoid local stress concentration. In other words, failure in practice of intricate shapes such as turbine blades might often occur, even though laboratory tests indicated that the strength of the material itself was adequate. Similarly, the low impact strength is an added danger, as materials which are liable to fly to pieces, rather than to bend, on collision with particles in the gas stream, clearly can not be tolerated.

Low ductility also contributes materially towards lowering the thermal shock resistance of ceramics. Low ductility would also appear to have a direct bearing on what may prove an overriding difficulty, even if ceramics can be developed which are otherwise entirely satisfactory, viz., the characteristic inconsistency or variance of properties from one sample to another—an effect found in all commercial refractories to some extent, but particularly in a material working close to its limits of strength. It is this factor which has so far limited the success of sintered alumina as a cutting tool for metals. While the precise causes of this inconsistency still remain to be established, it is perhaps reassuring to recall that the same difficulty was encountered, and eventually overcome, in the development of the cemented carbides.

Simulative tests show clearly that the thermal shock resistance of sintered oxide refractories (notably  $\text{Al}_2\text{O}_3$ ) is totally inadequate for the severe conditions of intermittent operation in gas turbines. The question whether the thermal shock resistance of such materials can be increased and to what extent is a problem of paramount importance and interest. The theory of elasticity

indicates clearly the factors which must govern this property in homogenous and perfectly elastic material. Under given conditions higher thermal shock resistance is conferred by higher mechanical strength and thermal conductivity and by lower modulus of elasticity and coefficient of expansion.

The overwhelming superiority of metals over ceramics as regards thermal shock resistance requires analysis. It is obviously due, in part, to the far greater thermal conductivity of metals. But their ductility, in strong contrast to the virtually complete rigidity of ceramics, must also be responsible. On this basis one approach to remedying the thermal shock deficiency is to attempt to introduce some measure of ductility into sintered oxide ceramics, and there appears to be two lines of attack: (a) by loosening the texture (for example, by under-sintering) so as to give individual crystallites some freedom to move relative to one another when suddenly stressed—the principle of stress accommodation which has been applied successfully in refractory practice; (b) by introducing a continuous interstitial phase of some ductile material such as a refractory metal.

As would be expected, in the case of (a), the elastic modulus is also reduced. At the moment it seems that some increase in thermal shock resistance might be obtained in this way, but it is most doubtful whether major improvements can be gained without losing the essential high strength of the material.

Recent work on sintered alumina, however, has shown that cross bend strength decreases less rapidly than Young's modulus as the sintering temperature of the material is reduced. Perhaps (b) is a promising line and the most important aspect of the ceramic-metal mixtures of recent introduction, which are now being actively investigated in many centres. Again, the question arises of compromising between gain in thermal shock resistance and loss of strength and creep properties.

Considerable information is available about the physical properties of many sintered oxide refractories, but many more data are needed before the potentialities can be fully assessed. Values given by different workers often vary very considerably because of differences in the method of preparation and testing and in the size of the test piece used. The data indicate however that the tensile strength of sintered oxides at room temperature may approach 10T/sq. in. All the evidence shows that the strength decreases progressively on heating but in certain cases (notably  $\text{Al}_2\text{O}_3$ ) it is still of a high order at 1,000°C.

The values for modulus of elasticity, strength, thermal expansion and conductivity do not point to any material, except BeO as likely to be particularly resistant to thermal shock. The outstanding and determining property in BeO appears to be the thermal conductivity which is stated to be exceptionally high. Very few data for creep strength of sintered oxides are as yet available, but the recent values for  $\text{Al}_2\text{O}_3$  in tension and bending at 1,000°C. appear to be promising.

Leaving aside the question of thermal shock resistance, it would appear from the relatively few data available that the sintered oxide ceramics already

offer considerable promise for turbine use. It appears that the materials have not been taken to their limit of development and that improvement in their properties, possibly substantial, may be achieved when the relation between constitution and properties is fully understood.

The problem of shock resistance is however serious, and at present precludes the use of sintered oxides for intermittently operated turbines. The prospects for successful application of ceramics seem much more promising with continuously operated turbines (for example, for stationary engines), in which the risk of thermal shock is presumably much less, or could be made so, by approximate adjustment of the operating conditions.

Multicomponent ceramics offer far more scope for reducing the thermal expansion coefficient by synthesis of crystalline or glass phases of intrinsically low expansion. The mineral cordierite, with an abnormally low coefficient of expansion  $1.0 \times 10^{-6}$ , is particularly interesting in this one respect although it is entirely ruled out for turbine use on the grounds of low refractoriness and creep at high temperatures. Faced with the need for developing low expansion refractories, ceramic specialists are now turning their attention to the fundamental problem of the relation between the expansion and the lattice structure of a crystal. This is an important line of research, but one which is not likely to yield practical results for some considerable time.

It has been found that certain bodies selected from this system could be vitrified in commercial kilns at 1,500-1,600°C. At the high firing temperature, however, creep under high stress was appreciable at 800°C. and pronounced at 980°C. A good composition as regards creep at 925°C. consisted of 80 per cent  $\text{ZrO}_2$ , 10 per cent  $\text{BeO}$  and 10 per cent  $\text{MgO}$ . Other bodies were interesting in having apparently good thermal shock resistance.

While much remains to be done in the field of multi-component ceramics, it already seems clear that such materials will be deficient in one respect or another, but particularly as regards strength and creep resistance at high temperatures, unless the component, and the compounds and eutectics formed from them, have exceedingly high melting points, for example, in the range 2,000-3,000°C.

The manufacture of such materials would necessitate firing at temperatures probably much higher than have yet been achieved commercially, while the starting materials themselves might be rare or difficult to obtain in bulk in the purity required.

# Some Recent Metallurgical Developments of Interest in Internal Combustion Engineering

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## Introduction

Mass production of internal combustion engine components which are subjected to severe service conditions—particularly alternating stresses at elevated temperatures—presents complex problems both in the choice of materials as well as in the methods of fabrication. Recent years have witnessed remarkable developments in both the above fields, and in fact, it has been said that the possibility of the internal combustion turbine and jet propulsion arises mainly out of the development of alloys capable of functioning at high temperatures.

## High Temperature Alloys

It may be admitted at once that the theoretical principles of alloy making have not advanced to that stage at which, given the requirements, one can prescribe a suitable composition and a thermal treatment. Constitutional diagrams of binary alloys are available of almost all the useful metallic elements, as well as a few ternary diagrams. These and the properties of the pure metals offer some guidance, but the majority of industrial alloys are complex systems and the behaviour of these under a set of conditions cannot be reliably forecast. The development of alloys, therefore, is largely empirical and their suitability has often to be decided by service tests. It has been said that “alloys and steels used in modern jet engines (which are of course only one application of the gas turbine engine) have to withstand stresses equivalent to hanging two saloon cars from a red hot piece of metal about the thickness of a fountain pen, without stretching or distorting it. They have, in fact, to withstand 10 tons/sq. in. at 850°C. Moreover, conditions in the engine are such that the fountain pen might have to be red hot at each end and comparatively cool at the centre. The necessity for meeting demands of this sort has led the steel industry's scientists into some unusual paths.”

A summary of the chemical composition of the more important high temperature alloys is given in Table 1.

## High Temperature Tests

The testing of these alloys includes high temperature creep and rupture tests. Some figures of the latter are given in Table 2 which is self-explanatory.

Since most of the components for use in internal combustion engines are to be manufactured to very close tolerances for interchangeability, as well as

TABLE 1—HEAT RESISTING ALLOYS

Type	Chemical Composition (%)						Remarks
	Ni	Cr	Mo	W	Cb	Ti	Other elements
1. Cr-Ni-Fe	8	18	0.4-3.0	1.3	0.3	0.2-1.5	Balance Fe
2. Ni-Cr-Fe	20-50	14-25	4-14	Up to 4	Up to 4.2		Balance Fe
Refractaloy A	50	20	14	—	—	—	Balance Fe
3. Ni-Cr-Co-Fe	15-33	14-27	3-10	Up to 4	Up to 4		Co, 13-45 N <sub>2</sub> , B, Ta- balance Fe
4. Ni-base: Hastel- loy B	65	—	29	—	—	—	Fe, 5
5. Ni-Cr & Ni-Cr-Co "Nimonic 80"	37-75 74	14-22 22	— —	— —	— —	— 2.5	Co up to 22 Al, 0.6
6. Co-Cr: Vitalium	—	28	5.6	—	—	—	Balance Co
7. Co-Cr-Ni	10-32	23-25	Up to 6	Up to 12	—	—	Balance Co

e.g., Timken  
Cr/Ni/Mo:  
16/25/6Forged  
buckets for  
use under  
1400°F.  
Hardened by  
Al or TiPrecision cast  
buckets

TABLE 2—STRESS-RUPTURE TESTS AT 1500°F.

Material	Stress for rupture in thousands of lb./sq. in. in hrs.		
	10	100	500
1. Cr-Ni-Fe ... ..	12·5-22	11-16	10-11
2. Ni-Cr-Fe ... ..	17-22·5	10·5-18	8·5-12·5
3. Ni-Cr-Co-Fe ... ..	20·5-34	16-25	12·5-20
4. Ni-base ... ..	25	15-17	11·7
5. Ni-Cr & Ni-Cr-Co-Fe Ti & Al hardened) ...	21-40	13-29	17-21
6. Co-Cr ... ..	21	11-28	14-25
7. Co-Cr-Ni ... ..	25-33	27-30	24-27

for considerations of high operating efficiency, dimensional changes under working conditions and during the working life of the engine have to be known beforehand. Creep tests are used for this assessment which measure the deformation in terms of the stress in lb./sq. in. required at a particular temperature to cause a creep rate of one-ten-millionth of an inch per hr. for every inch of the length of the specimen. The figures given in Table 3 are indicative only of the magnitude of the creep.

TABLE 3—SOME CREEP DATA OF HIGH TEMPERATURE ALLOYS  
(TEMP. 1350°F. ; STRESS 15,000 LB./SQ. IN.)

Material and Treatment	Creep rate per hr. @ 1000 hrs.
1. Cr-Ni-Fe: C, 0·26 ; Cr, 0·18 ; Ni, 0·8 (Oil-quenched and aged)	0·00001
2. Ni-Cr-Fe: Co, 0·4 ; Cr, 0·13 ; Ni, 0·19 ; Mo, 4·28 W, 3·87 ; Co, 4·2 (Water-quenched and aged)	0·000101
3. Ni-Cr-Co-Fe: Refractaloy M284 C, 0·11 ; Cr, 0·20 ; Ni, 0·20 ; Co, 0·30 Mo, 0·8 ; W, 0·4 (Oil-quenched and aged)	0·000052
4. Ni-Cr-Co-Fe: (Ti and Al hardened) Refractaloy 26 C, 0·03 ; Cr, 0·18 ; Ni, 0·37 ; Co, 0·20 Mo, 0·3 ; Ti, 3 ; Al, 0·25 (Oil-quenched and aged)	0·00001
5. Co-Cr:—Vitallium C, 0·2 ; Cr, 0·28 ; Mo, 5·6 ; Co, bal. (Cast and aged)	0·000040

Similar data for other values of stresses and duration have been investigated and reported in the literature, on the basis of which design data have been arrived at, which are helpful in deciding the suitability of an alloy for a particular application.

### Newer Fabrication Methods

Under the newer fabrication techniques particular mention may be made of the centrifugal and precision casting methods.

In centrifugal casting, the molten metal is poured into the centre of a rotating mould, thus forcing the metal into the cavities. This mould is made up of a number of segments assembled in the form of a stack, and provided with proper runners and gates.

Precision casting is not really a new method, since it is only a highly developed form of the well known "lost wax" process. This process has assumed considerable importance now because some of the high temperature alloys are not forgeable and are of poor machinability. For their use the castings themselves should practically conform to the dimensions required in the assembly line.

The method consists of making the required shape accurately in the form of a wax-casting using a master steel mould. A number of these wax castings are assembled in a mould box, and a refractory, air-setting investment of syrupy or thicker consistency, is poured round the assembly. After the investment has set, the moulds are gradually heated to melt and burn off the wax assemblies, leaving the corresponding cavities in the mould. These hot investment moulds are mounted on a centrifugal casting machine, and an alloy of a suitable composition is poured into the rotating mould. After cooling, the mould is broken off, and the castings are sand blasted lightly. A few grinding operations may be necessary before the castings can be used in the assembly line.

### Conclusion

It will be clear from the preceding account that metallurgical developments have a profound influence in the design and manufacture of internal combustion engines.

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# High Duty Materials for Engine Components

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## 1. Introduction

This report deals briefly with materials commonly used for various components of an internal combustion engine and on this basis certain conclusions are drawn regarding the availability of the same in India and action to be taken in this direction.

## 2. Choice of Materials

The choice of materials for various components of an internal combustion engine depends to a very large extent on the type of the engine i.e., petrol or diesel, and on the purpose for which it is intended. For this purpose, all engines can be broadly divided into three groups: (a) Industrial engines, (b) Transport engines and (c) Aero engines.

To-day it can be said that diesel engines, by virtue of use of heavier distillates of minerals oils and low specific fuel consumption, monopolise the whole field of industrial engines. For the same reasons, diesel engines are being used in large numbers for transport purposes and in this field also they are gradually wresting the initiative from petrol engines.

Petrol engine, by virtue of its low weight, monopolises the field of aero engines (piston engines).

So, when dealing with industrial and transport engines, only materials used for diesel engines will be mentioned. In the case of Aero engines, materials used for aircraft petrol engines will be mentioned.

In aero engines low weight is of primary importance and cost is of secondary importance. That is why, petrol engines are mostly used for aero engines. So materials chosen for these engines should fulfill the following conditions: (a) Low specific weight and (b) High strength and good fatigue resisting properties. High grade materials are used and a low factor of safety is generally adopted.

Since for industrial engines, generally, low cost is the primary consideration and weight is of secondary importance, the best among the comparatively cheap materials are used and a high factor of safety is usually allowed.

In transport engines all efforts are directed to obtain a low weight with minimum cost and so high grade materials which fulfill these conditions are used.

## 3. Components and Materials Used

When choosing material for any component of the engine, the function of the component and the conditions under which it operates have also to be considered. Here the main components of the engine will be taken up one

by one. Their functions and materials used will be briefly mentioned in the cases of (a) industrial engines (b) transport engines and (c) aero engines—wherever there is a marked difference.

3.1. CRANKSHAFT—An engine crankshaft has to fulfill the following conditions in order to perform its functions properly: (a) Enough strength to withstand the forces to which it is subjected ; (b) enough rigidity to keep the distortion a minimum ; (c) sufficient mass, properly distributed to see that it does not vibrate critically at the speeds at which it is operated ; (d) sufficient bearing area to keep down the bearing pressure to a value dependent on the lubricant available and (e) minimum weight, especially in aero engines.

To meet the rigidity and vibration requirements it is usually necessary to make the shaft much heavier and stronger than would be necessary from the strength point of view. Hence, it is not possible to reduce the weight of the crankshaft appreciably by using a material having a very high strength.

The material for crankshaft also depends on the method by which it is produced, i.e., cast, forged or built-up. Built-up crankshafts are sometimes used in the case of aero engines where, as already mentioned, low weight is of specific importance.

In industrial engines, 0.35 carbon steel of ultimate tensile strength (UTS) 32/38 tons/sq. in. and 0.45 carbon steel having UTS 40/50 tons/sq. in. are commonly used.

In transport engines, alloy steels, for example, manganese steel having UTS 50/60 tons/sq. in., are commonly used.

In aero engines, nickel chromium steel having UTS 60/70 tons/sq. in. is generally used.

Heavy duty cast iron is being successfully used for crankshafts, especially in industrial engines of comparatively low speeds. Nodular cast iron—UTS 35-44 tons/sq. in., acicular cast iron UTS 25-40 tons/sq. in. and Meehanite UTS 26-35 tons/sq. in. are specific examples of heavy duty cast iron which have good mechanical properties.

In the I.C.E. Department work has already been initiated in this direction and a heavy duty cast iron crankshaft has been fabricated for an experimental engine. It has acicular structure and tests on specimen bars have given very favourable results.

Cast iron crankshafts can replace easily 0.35-0.45 carbon steel crankshafts used in industrial engines. This will go a long way in reducing the cost of the engine.

It should also be mentioned here that cast steel is also used as material for crankshafts.

3.2. CONNECTING ROD—The connecting rod should have adequate strength and stiffness with minimum weight.

In industrial engines carbon steel having UTS 35-40 tons/sq. in. is commonly used. In transport engines alloy steels having a strength of 50-60 tons/sq. in., e.g., manganese steel, is commonly used. In aero engines, nickel chrome steel having UTS 60-90 tons/sq. in. is most commonly used.

3.3. BEARINGS—In general the choice of material for bearing depends

on the load coming on it and the rotational speed of the engine. Where the loads are not too great, bronze or steel backed, white metal lined bearings are used. Bearings with copper-lead or cadmium alloy lining are used where the bearing pressures are high. In some cases bearings with a flashing of white metal are used.

3.31. *Main bearings and connecting rod big-end bearings*—In low and medium speed diesel engines 0.15-0.25 carbon steel backed, white metal lined bearings are commonly used. In high speed diesel engines 0.15-0.25 carbon steel backed, lead bronze lined bearings are usually used. In aero engines, bearings with thin low carbon steel shells lined with white metal are common. Ball and Roller bearings are sometimes used for main bearings in high speed engines, especially in aero engines.

3.32. *Connecting rod small-end bearings*—Phosphor bronze bushes are very commonly used for small end bearings. In some high speed diesel engines needle bearings are used.

3.4. **PISTON**—An engine piston should fulfil the following requirements: (a) low coefficient of expansion, (b) high coefficient of thermal conductivity, (c) low density, (d) high resistance to abrasion and (e) adequate mechanical strength at working temperatures.

Common materials used for pistons are cast iron and aluminium alloy. Neither of them is satisfactory from every stand point. In engines where piston speeds are below about 1,200 ft./min. (approx.) cast iron is usually used for pistons. Above the piston speed of 1,200 ft./min. (approx.) aluminium alloy is usually used. In high speed diesel engines and aero engines, silicon aluminium alloy having an UTS 14 tons/sq. in. is usually used for pistons.

3.5. **PISTON RINGS**—Piston rings should be sufficiently elastic to exert the necessary side pressure against the cylinder walls and to permit insertion of the ring in its groove by sliding it over the piston. It should be relatively soft to prevent excessive wear on the cylinder walls.

Pearlitic cast iron is almost universally used for piston rings. Sometimes rings are chromium plated in order to reduce wear and thereby increase the life of the rings.

3.6. **PISTON PINS**—Materials used for piston pins should have high strength and surface hardness. Plain carbon case-hardening steel and nickel steel are commonly used in all types of engines. Nickel steel having UTS of 80 tons/sq. in. is most commonly used for piston pins of aero engines.

3.7. **VALVES AND OPERATING GEAR**—Valves, especially exhaust valves, should have (a) high strength at high working temperature (b) maximum resistance to distortion or warping (c) sufficient hardness and resistance to impact to prevent wear (d) resistance to corrosion and oxidation and (e) no tendency to air-harden when cooled rapidly.

For exhaust valves silicon chromium steel and nickel chromium steel are commonly used.

In diesel engines carbon steel having an UTS 45-55 tons/sq. in. is used generally, for inlet valves.

In aero engines, silicon chromium steel and nickel chromium steel are commonly used for inlet and exhaust valves. Usually valve seats, valve contact cones and tips are stellite.

In diesel engines, especially for exhaust valves, medium cast steel inserts are used for valve seats. Swedish steel is mostly used for valve springs. For valve push rods high carbon steel is generally used.

Low carbon case-hardening steel and nickel steel are the materials generally used for camshafts. Cast iron camshafts are now-a-days very popular. For camshaft bearings, generally solid bronze bushes are used.

3.8. CYLINDER HEAD, CYLINDER BLOCK AND CYLINDER LINER—In diesel engines cast iron having UTS 16.5 tons/sq. in. is generally used for cylinder head and cylinder block. In aero engines cylinder heads are usually made out of aluminium alloys.

Both wet and dry liners find equal application in diesel engines. The material generally used is cast iron either alloyed or unalloyed, having UTS 16.5 tons/sq. in. In aero engines cylinder barrels are generally machined from 0.5-0.6 carbon steel forging.

3.9. CRANKCASE—In low and medium speed diesel engines cast iron having about UTS 16.5 tons/sq. in. is usually used for crankcase. Sometimes cylinder block, crankcase and engine base are made, integral in one piece, from cast iron. In some high speed diesel engines crankcases are welded steel structures.

In aero engines crankcase and covers are aluminium alloy castings.

3.10. BOLTS AND NUTS—High tensile steel bolts are generally used in internal combustion engines. The material for this is generally manganese steel or chromium steel. The material for nuts is usually mild steel.

#### 4. Review

On the basis of the above analysis, the position, regarding availability of materials in India for various components of an internal combustion engine can be studied.

Generally, about 90 per cent by weight of a diesel engine is cast iron. Steel components form about 7 per cent of the weight of the engine and the remaining are non-ferrous materials. In aero engines steel components and non-ferrous materials form a higher percentage. From this it can be seen that the major constituent in the materials for a diesel engine is cast iron.

4.1. CAST IRON—The basic material for cast iron i.e., pig iron, is being produced in India by Messrs. *Tata Iron & Steel Co.*, *Steel Corporation of Bengal* and *Mysore Iron & Steel Works*. Cast iron required for I.C. engines has to be produced from this basic raw material by proper addition of alloying elements. This requires plenty of development work.

In this country, Messrs. *Cooper Engineering* and their associates are making 'Meehanite', a patented heavy duty cast iron, which is suitable for engine castings like cylinder head, cylinder block, crankshaft, etc.

4.2. STEEL COMPONENTS—As already mentioned, steel components form a small percentage by weight of the engine. Since 1938, Messrs. *Tata Iron*

& Steel Co. are producing in small quantities carbon steels and alloy steels which are quite suitable for most of the engine components using steel as the material.

#### 4.3. Non-ferrous Materials

4.31. (NON-FERROUS MATERIALS) *Aluminium*—Messrs. *Aluminium Corporation of India* and *Indian Aluminium Co.* produce aluminium in this country. From this, by proper addition of alloying elements, aluminium alloy suitable for engine components have to be developed.

4.32. *Bearing materials*—Only in the case of bearing materials, which form a very low percentage of the weight of the engine, it may be necessary at present to import basic raw materials like tin, copper, lead, etc. Using these raw materials, materials suitable for engine bearings have to be developed.

#### Conclusions

Even though most of the raw materials, necessary for producing alloys required by the internal combustion engine industry, are available in India, firms in India are not at present able to supply the alloys correct to desired specifications. This is due to the fact that there is no proper demand and so it is not economical for the firms to conduct research on the production of these alloys according to desired specifications. Somewhere this vicious circle of demand and supply has to be broken for successfully solving the problem of materials for I. C. engines.

In foreign countries, especially in England, due to scarcity of alloying materials like nickel, chromium, etc., low alloy or plain carbon case-hardening steels, which have mechanical properties similar to that of the high grade alloys, have been developed. Efforts to produce heavy duty cast iron have resulted in nodular and acicular cast irons having UTS 35-44 tons/sq. in. and 25-40 tons/sq. in. respectively. Here in India too, similar development work to produce alternative materials to high grade alloys, will have to be carried out.

As a solution to the problem of materials for I. C. engines, it is suggested that development of materials and design and development of engines be carried out simultaneously so that this will result finally in engines which can be mass produced from indigenous materials.

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# Testing and Design of Air Cleaners

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## 1. Introduction

Atmospheric air contains suspensions of finely divided particles such as dust, fog, mist, fumes, smoke, etc. Such particles enter an engine with the air breathed in. Dust particles contain hard siliceous matter in considerable proportions and thus cause the rapid wear of the different parts of the engine. Therefore, various types of air cleaning devices have been developed to remove the dust particles from the air before it enters the engine.

The performance of air cleaners depends on the pressure drop, general efficiency and separating property for the particular particle size range. Testing of an air cleaner for its performance consists in finding out quantitative data in respect of these qualities.

In the following some data regarding dust particles and their effect on internal combustion engines are given.

The composition, concentration and size of dust particles in the air varies from place to place. Clean country air contains as little as 0.1 to 0.2 mg. of dust per cubic meter and a considerable proportion of the dust is organic matter. The concentration might go up to 500 mg./cu.m. in industrial areas. The size of dust particles may be taken as  $0.5 \mu$  ( $1 \mu = 0.001 \text{ mm.}$ ) for ordinary conditions. The size may be  $100 \mu$  or more under disturbed atmospheric conditions and industrial areas. Fig. 1 shows the sizes of the different types of suspensions in the air. The effect of dust particles on different types of engines are reviewed.<sup>1,5-9</sup> The life of different engine parts such as cylinder, piston, piston rings and valves is reduced considerably, in different proportions, depending on the conditions of the air. Naturally, the effects are severe with tractor engines, aircraft engines working in desert regions and other internal combustion engines working in industrial establishments without a suitable air cleaning device.

Numerous air cleaning devices have been developed for the purpose of eliminating dust particles.<sup>5,6,8,10</sup> All these devices work on the principle of filter and/or inertia and/or centrifugal separation.

## 2. Performance of an Air Cleaner

The output of an internal combustion engine drops with an air cleaner having a high pressure drop: The collection efficiency of an air cleaner depends on its separation quality. This, of course, depends on the type of air cleaner. The separating quality of an air cleaner with regard to its efficiency with particles of different size ranges is also very important.

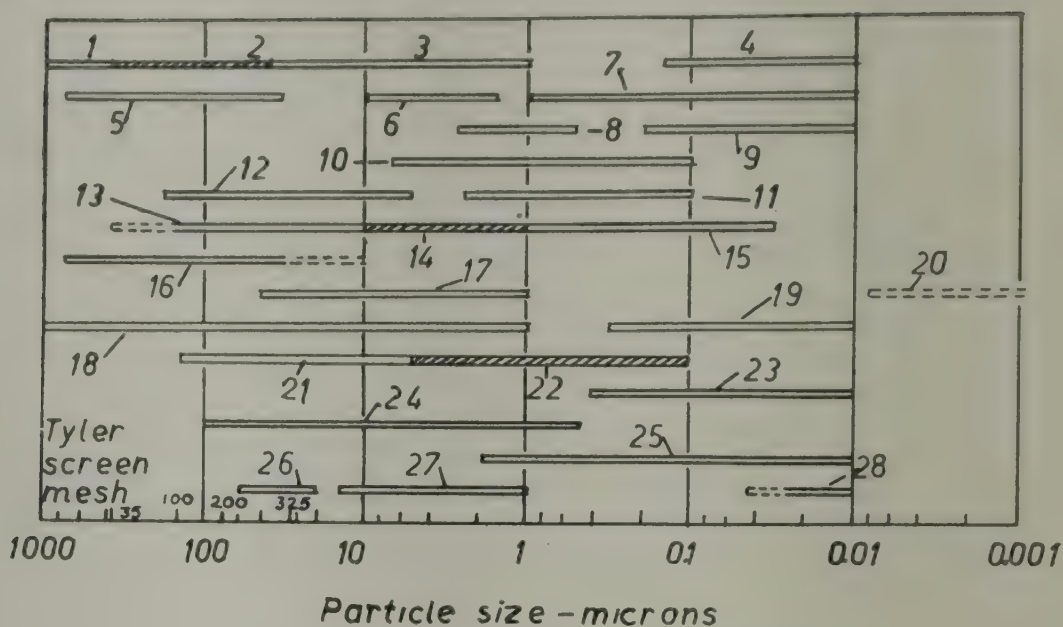


FIG. 1—PARTICLE SIZE RANGES FOR AEROSOLS, DUSTS AND FUMES. (1. Rain drops ; 2. Mist ; 3. Fog ; 4. Tobacco smoke ; 5. Ground limestone ; 6.  $\text{H}_2\text{SO}_4$  concentrator mist ; 7. Rosin smoke ; 8.  $\text{SO}_3$  mist ; 9. Carbon black ; 10. Pigments ; 11.  $\text{NH}_4\text{Cl}$  fume ; 12. Sulfide ore for flotation ; 13. Pulverised coal ; 14. Spray dried milk ; 15. Oil smoke ; 16. Stoker fly ash ; 17. Pulverised coal fly ash ; 18. Foundry dusts ; 19. Zinc oxide fume ; 20. Diameter of gas molecules ; 21. Cement dust ; 22. Alkali fume ; 23. Magnesium oxide smoke ; 24. Metallurgical dust ; 25. Metallurgical fume ; 26. Pollens ; 27. Bacteria ; and 28. Virus & Protein.

Recent tests with R.1830 Twin Wasp single cylinder engine have proved that other factors being equal, the wear produced by a given weight of silica dust depends on the size of the dust. The maximum wear occurs with a dust size of  $15\ \mu$  wear with  $100\ \mu$  dust being approximately half that of  $15\ \mu$ .

Tests on a light tank engine made by the Cadillac motor car division of the General Motor Corporation with two size ranges of dusts 0 to  $2.5\ \mu$  and 0 to  $40\ \mu$  showed an increase in wear of 50 per cent with coarser dust. Similarly Allis Chalmers made tests on four cylinder engines fitted with cast iron cylinders, pistons and piston rings, feeding three dusts of  $0.5\ \mu$ , 5 to  $15\ \mu$  and 15 to  $30\ \mu$  size range at a concentration of 25 mg./cu. ft. They found that 5 to  $15\ \mu$  dust produced one and a half times as much wear as the  $0.5\ \mu$  and 15 to  $30\ \mu$  dust four times as much. The testing of an air cleaner for separating properties on different size ranges is therefore essential. Perhaps, an air cleaner showing higher efficiency with coarser size of dust particles might have a very low efficiency with the finer dust.

### 3. Technique of testing Air Cleaners

The testing apparatus should be able to give complete data on the performance of the cleaner under the required conditions. Pressure drop across the air cleaner can be measured directly for a particular mass flow by allowing

a measured amount of air to pass through the cleaner and finding the pressure drop by means of a suitable instrument. Dust of known size may be uniformly supplied by means of a suitable mechanism to a measured amount of air. The air and dust should be mixed thoroughly before they enter the air cleaner.

The dust which is not filtered, by the testing air cleaner, may be filtered by means of an absolute filter and weighed. This will check up the collection efficiency if the dust separated by the air cleaner could be weighed. This is necessary to determine the collection efficiency when the dust separated cannot be weighed.

The use of a high power microscope with a measuring device is helpful in determining the performance of an air cleaner. By knowing the characteristics of dust leaving the air cleaner, by means of the microscope, it is possible to determine the filtration power with different particle sizes. Accuracy will depend upon the accurate sampling of dust for examination under the microscope. To know the separating properties of an air cleaner for different size ranges of dust particles, dust should first be sampled into required size ranges and then fed to the dust feeder.

Fig. 2 shows the diagrammatic sketch of the apparatus used to test the new designs of cyclonic type of air cleaners. The supply of air is controlled on the delivery side by means of the gate valve *G* and it is measured by means of an orifice meter *O* at the suction side. A blower *B* is used to supply the required air mass flow. *D* is the dust feeder which is used to feed the dust into the compressed air. This is just similar to that developed by Fisher and Davis.<sup>12</sup> The vibrator for the feeder consists of a small d.c. motor  $1/20$  h.p., 220 volts, 2,200 r.p.m. and steam engine like mechanism and links.

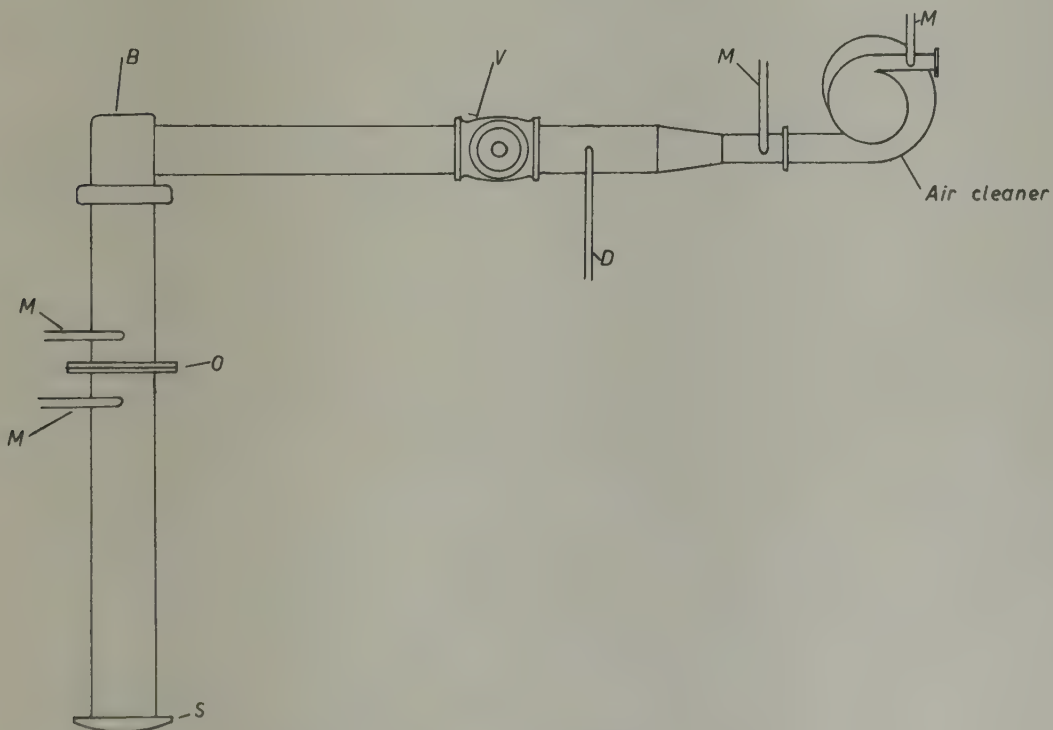


FIG. 2—DIAGRAM OF AIR FILTER TEST-RIG

It is possible to control the feed of dust from the dust feeder by varying the speed of the motor and thus the vibrations of the feeder. Thus a known quantity of dust is fed at uniform rate to the measured supply of air. The feeder is closed air tight at the top to equalise the pressure in the feeder. The manometers for recording the pressure drop are indicated by MM. The test dust selected is fine porcelain powder. It is sampled into the following size ranges by means of the standard Tyler sieves.

Size range ( $\mu$ )				%
0-44	...	...	...	19
44-74	...	...	...	30.7
74-104	...	...	...	34.6
104-147	...	...	...	14.1
above 147	...	...	...	1.51

The dust is dried in a dryer before feeding it into the system.

#### 4. Improvements in the Performance of cyclonic type of Air Cleaners

Of all the dust separating devices, the cyclone separator is the most simple in operation. It does not require frequent servicing. If properly designed a high separating efficiency can be expected.

But cyclones suffer from numerous disadvantages. The main disadvantage is their high pressure drop. In the cyclone type of gas cleaning and separating devices, a swirl motion is imparted to the gas flow and the suspended particles, mostly solid, are consequently subjected to a radial acceleration and are carried away to the wall of the separator from where they are removed. In most of the cases the clean gas generally escapes from the chamber through an axial tube which terminates below the entrance section of the cyclone. There is considerable pressure drop due to the sharp reversal in the direction of flow of the gas inside the separator, thus affecting the engine output if the separator is a part of the power plant.

Moreover, the outlet tube of the separator which is usually made of metal is protruding inside the body of the separator. The result is, that in cases of application of the separator to the thermal power plants and especially turbines run on fuel of high residual contents, this part gets unduly heated due to the high temperature of the gas stream and is rendered liable to deformation cracking.

It has been reported that the spinning kinetic energy of the outgoing clean gas is the main cause for the high pressure drop of the cyclone. C. J. Stairmand<sup>15</sup> has deduced an energy equation for the pressure drop in a cyclone separator. He gives total loss of head = centrifugal head in cyclone vortex + kinetic head at inlet + loss at exit - kinetic head recoverable from the spinning gases leaving the cyclone. Hence, an appreciable reduction in the pressure drop can be effected if the loss at the exit is reduced and some energy

for the spinning gas which leaves the cyclone is recovered. He has given the equation:

$$\text{Pressure loss} = C \left[ V_I^2 \left\{ 1 + 2\phi^2 \left( \frac{R_I}{R_E} - 1 \right) \right\} + 2V_E^2 \right]$$

where  $V_I$  = linear speed in inlet duct

$V_E$  = „ „ exit „ „

$R_I$  = radius of circle to which centre line of the inlet is tangential

$R_E$  = radius of the exit pipe

$\phi$  = velocity ratio  $\frac{U_I}{V_I}$  where  $U_I$  = spinning speed at mean inlet radius and

$C$  = a constant which depends on the units

excluding the kinetic head recoverable. So it can be seen that the geometry of the cyclone also is a factor in the reduction of pressure loss in a cyclone.

Various types of devices have been developed for the reduction of pressure drop. However, few of them are of practical simplicity. A two cone system for the outlet has given reduction up to 34 per cent. The straightening vanes have given a static pressure recovery of 24 per cent with four vanes at an angle of 72 per cent with the outlet cone. These lead to the conclusion that there is a possibility of reducing the pressure drop of a cyclone separator either by improving the geometry of the cyclone or recovering the spinning kinetic energy from the outgoing clean gas.

The other main factor in the performance of an air cleaner is its collection efficiency. It has been shown that for geometrically similar cyclones, the cyclone of smaller diameter has higher efficiency, and small cyclones are better for removing very small particles from the air. Lissman<sup>20</sup> has shown that while in a 10 ft. diam. cyclone the force on a certain size of particle in the outer vortex was 22 times the force of gravity, the force in a 4 in. diam. cyclone was 672 times gravity for the same particle with the same gas velocity.

Again, the study of flow pattern in a cyclone separator shows that it consists primarily of an outer downward spiral and inner upward spiral of higher velocity. Theoretical analysis confirmed by experimental results has shown that in the case of a cyclone the value  $U_R^2 \cdot R$  is constant so that at any radius,

$$U_R = U_{R_1} \sqrt{\frac{R_1}{R}}$$

where  $U_R$  = spinning speed at radius  $R$

$U_{R_1}$  = „ „ „ „  $R_1$

This relation is true for the outer spiral of the flow pattern. Hence, the region near the separator wall is the region of least velocity. It has also been shown theoretically that the smaller dust particles have a tendency to rotate at smaller radii. If by some suitably shaped insertion the particles which are separated are led towards the periphery or the outer wall of the body of the separator by centrifugal action assisted by a suitably shaped insertion, a higher efficiency can be expected.

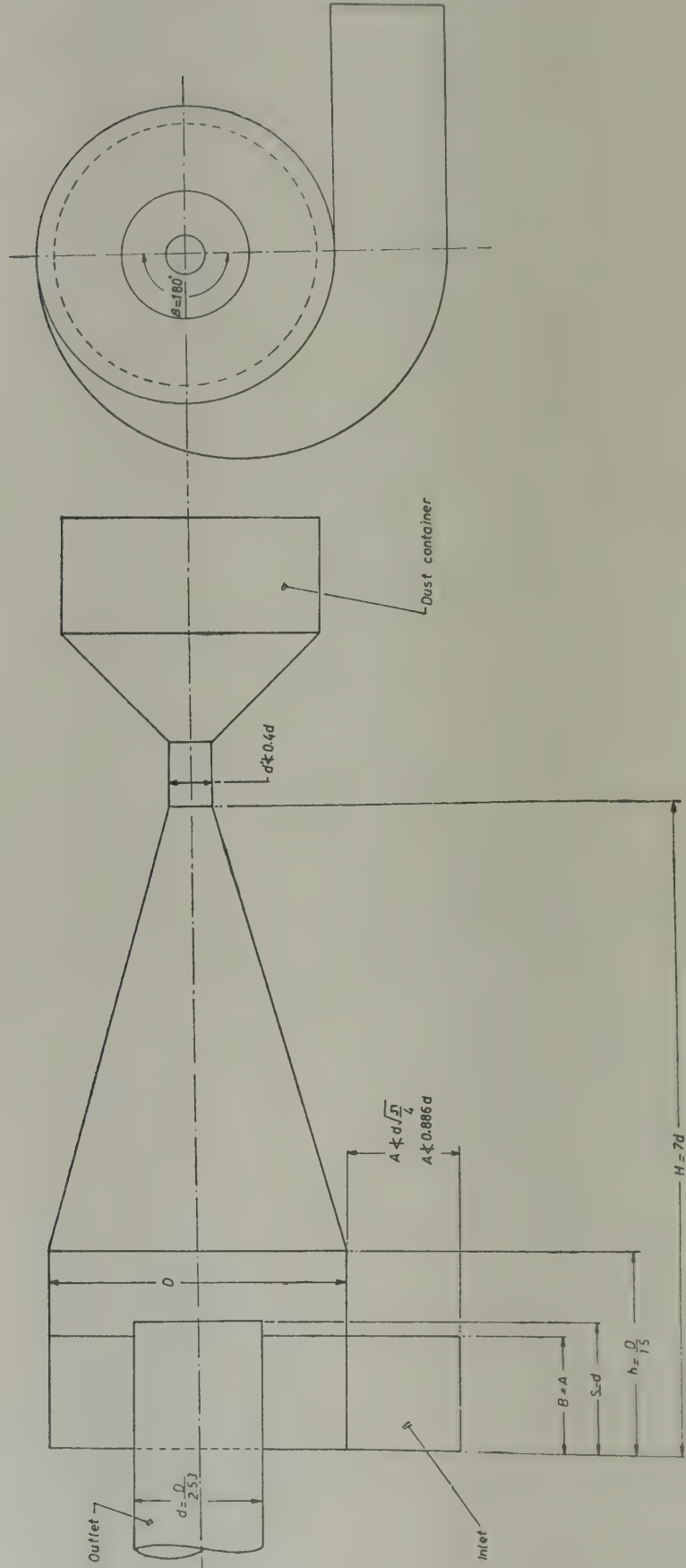


FIG. 3—SEPARATOR AS OUTLINED BY TER LINDEN

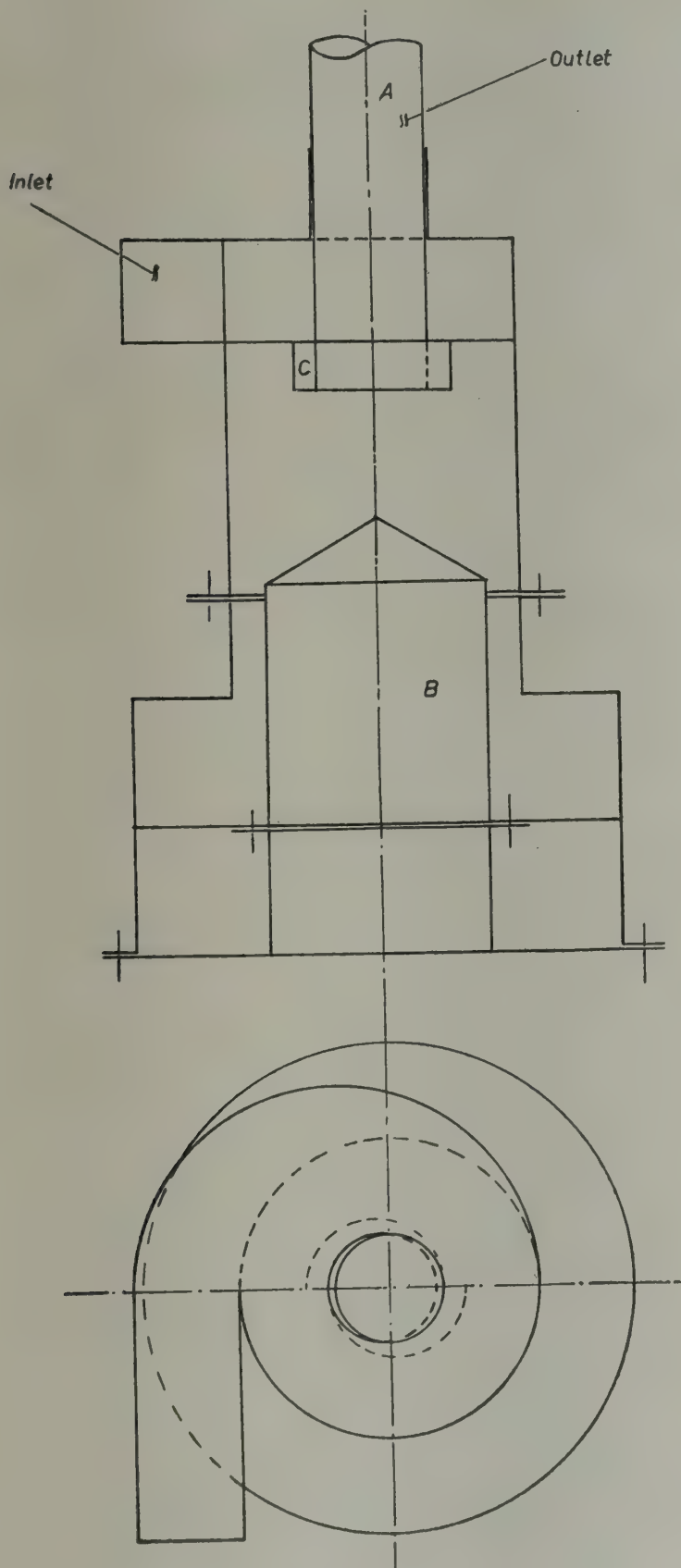


FIG. 4

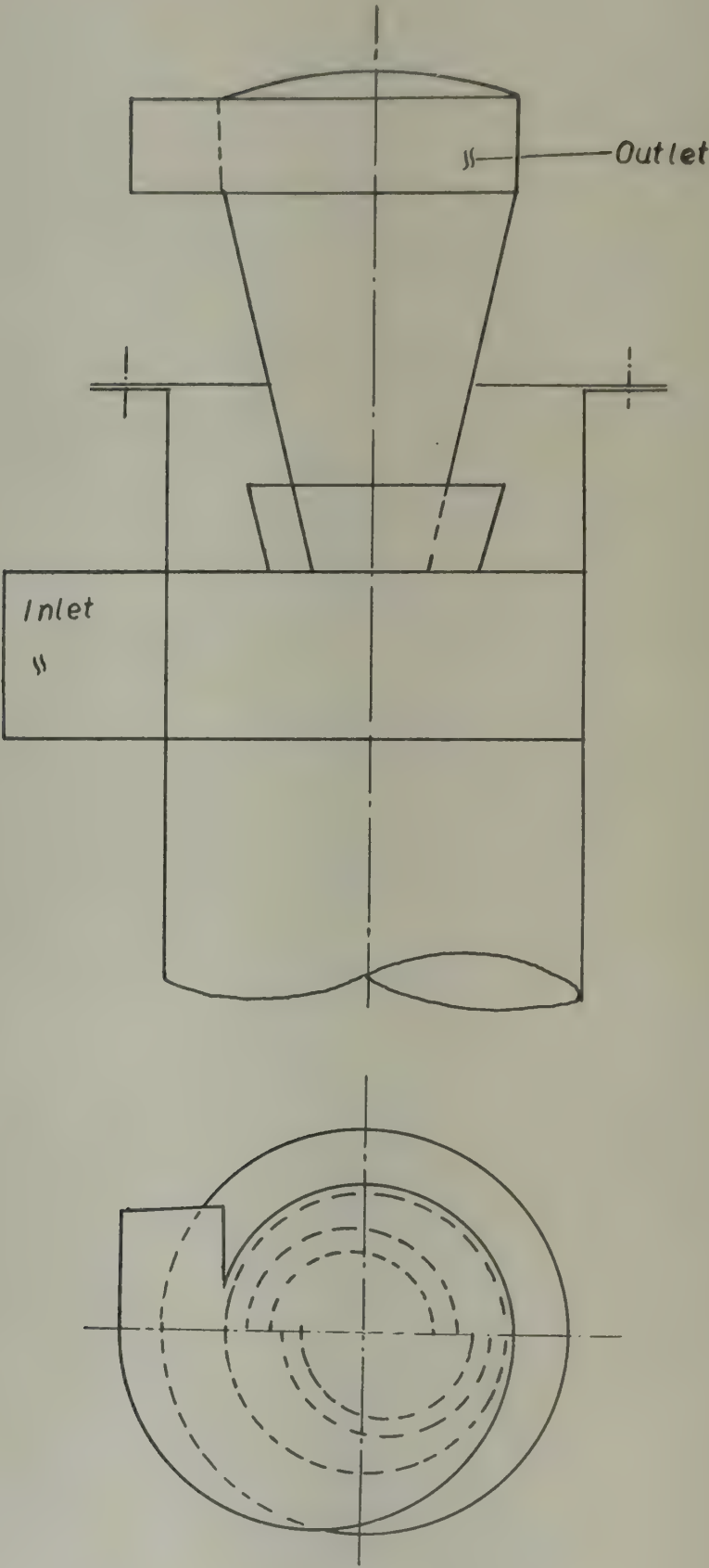
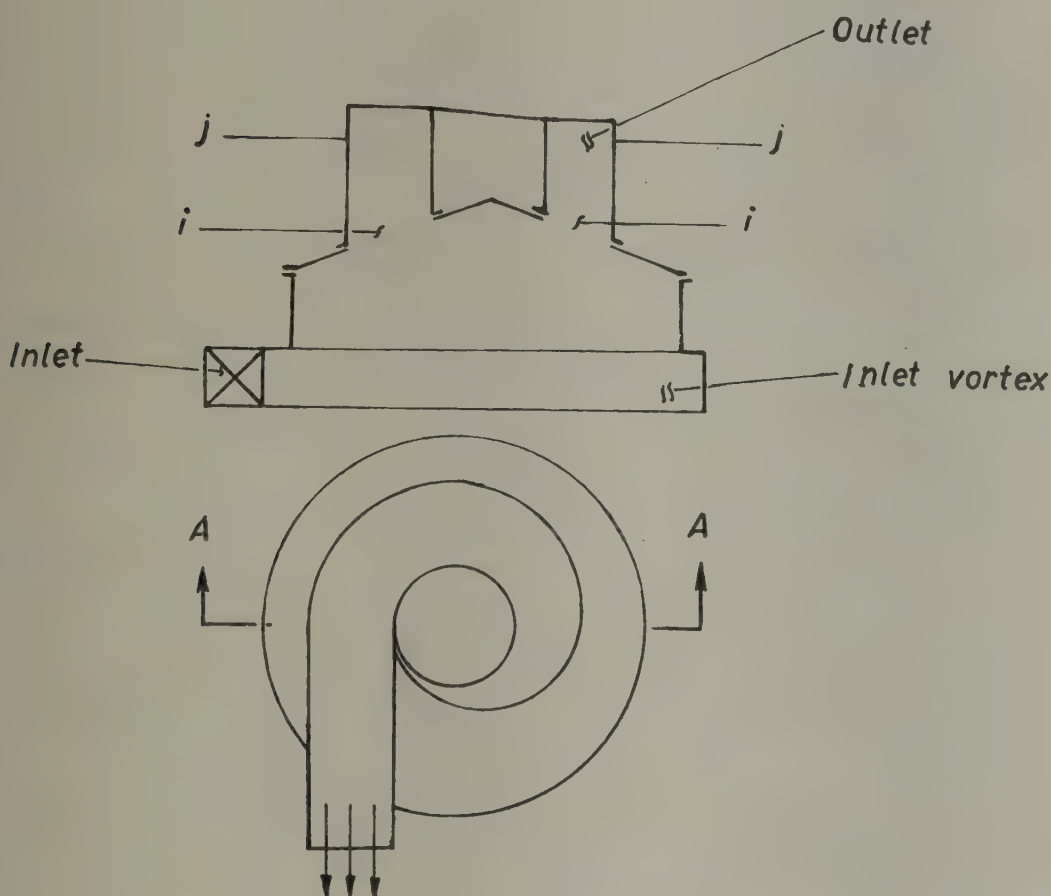


FIG. 5

Fig. 3 gives the conventional type of separator as outlined by Ter Linden<sup>21</sup> which was used for comparison. Figs. 4, 5 and 6 give the modifications introduced for improvement in the performance of cyclonic type air cleaners. In Fig. 4, a conical insertion has been used in order to effect higher collection efficiency. The outlet tube has got one or two tangential slots. The bottom of the outlet tube might be open or closed. The effective position has to be determined by actual experiments. Fig. 5 shows the conical outlet tube with longitudinal slots and with a spiral outlet tube. All the devices aim at the recovery of spinning kinetic energy from the outgoing gas and thus reduce the pressure drop.



FIGS. 4, 5 & 6—MODIFICATIONS INTRODUCED FOR IMPROVEMENT IN THE PERFORMANCE OF CYCLONIC TYPE AIR CLEANERS

All the models are to be tested with different cone angles for this performance. The size of the cone also is to be varied. It may be hoped that a new model with suitable modifications may give better performance than the conventional type. A new apparatus for sampling dust into different size ranges is proposed to be developed.

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# The Use of Simple Epicyclic Gears

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To reduce the speed of engines or electric motors an epicyclic gear may be used with the following advantages: (1) coaxial in- and output shafts, (2) short and round gear box, (3) small number of gears for high ratios of transmission, (4) good degree of efficiency by use of internal gears, (5) easy disengagement or change of speed.

## Two-Gear Trains

A two-gear train with fixed gear carrier C (Fig. 1) becomes an epicyclic two-gear train by fixing gear 1 instead of carrier C (Fig. 2). By exchanging one of the external gears in Fig. 2 against an internal gear we get gear train (Fig. 3 or Fig. 4). The vectors show the velocities, the angles  $\beta$  according to  $\omega = \text{const.} \times \tan \beta$ , the angular velocities. For a coaxial second shaft  $o$  to the shaft  $i$  we may use a parallel drive in Figs. 3 and 4 which drive is not recommended in Fig. 2 because carrier C does not allow a simple design.

In Fig. 3 four pivots A are mounted on the planet gear 2 at a distance 'a' from the rotating centre  $C_2$  of gear 2. On shaft O, a star with four pivots B is fixed, the pivots having the same distance 'a' but from the centre  $C_1$  of the rigid sun gear 1. Pivots A are connected with the pivots B by couplings with the length  $C_1C_2 = b$ , thus getting parallelogram  $C_1C_2AB$ .

Using the carrier shaft as input shaft with the angular input velocity  $\omega_c$ , centre  $C_2$  has the velocity  $v_c = \omega_c b$  (Fig. 3). With the instantaneous centre P for gear 2 and by rotating the velocity-vectors through  $90^\circ$ , we find  $v'_c$ , then  $v'_a$  and  $v'_b$  by parallel lines. Rotating back we get vector  $v_b$  in its true position, indicating the direction of rotation of output shaft O opposite to the direction of rotation of the input shaft  $i$ . From output velocity  $-\omega = v_b/a$ , the relations  $v'_b : v'_c = AC_2 : PC_2 = a : r_2$  and  $v_c = \omega_c b = \omega_c (r_1 - r_2)$ , where  $r_1, r_2$  denote the pitch radii of gear 1 and 2, we find

$$\omega = \omega_c \frac{r_2 - r_1}{r_2} = \omega_c \frac{t_2 - t_1}{t_2} \quad \dots \quad (1)$$

with  $t_1, t_2$  as the numbers of teeth. The limit of the difference  $t_2 - t_1$  is given by the addenda and a clearance between the addendum circles allowing the teeth to come out of contact. With the addendum equal to  $\frac{1}{p}$  with  $p$  as diametral pitch and also the clearance equal to  $\frac{1}{p}$ , it may be assumed

$$2r_1 - 2r_2 = \frac{2}{p} + \frac{1}{p} = \frac{3}{p} \quad \text{or, with } 2r_1 = t_1/p, t_1 - t_2 = 3 \text{ as the limit.}$$

Fig. 1

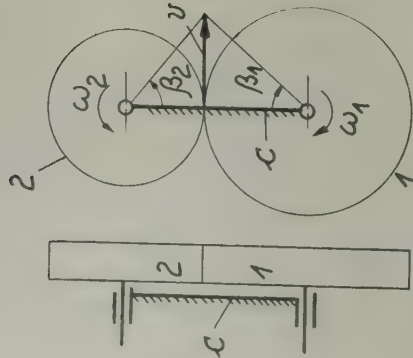


Fig. 2

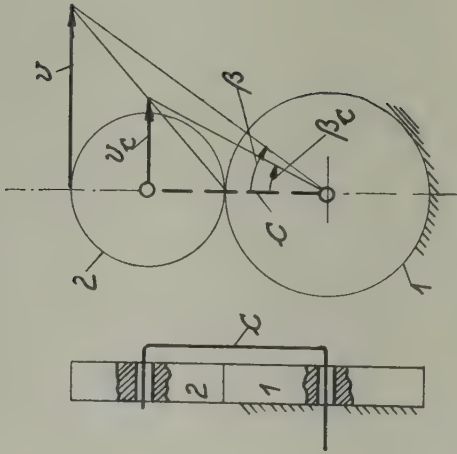


Fig. 3

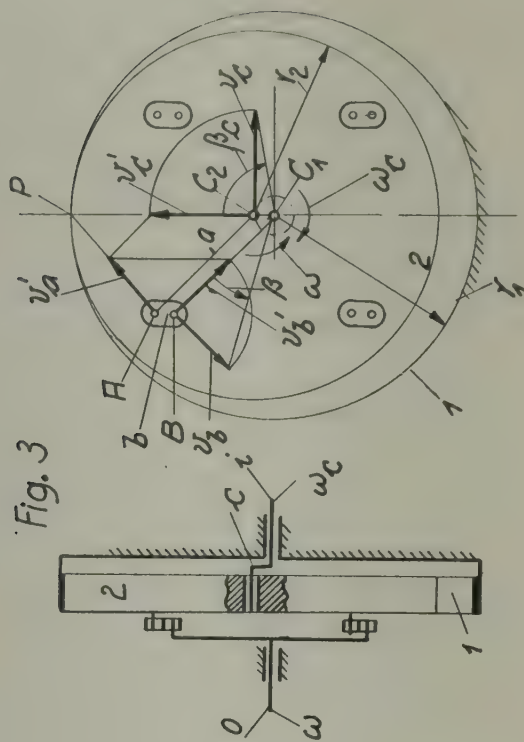


Fig. 4

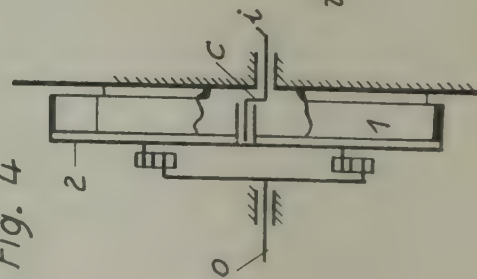


Fig. 5

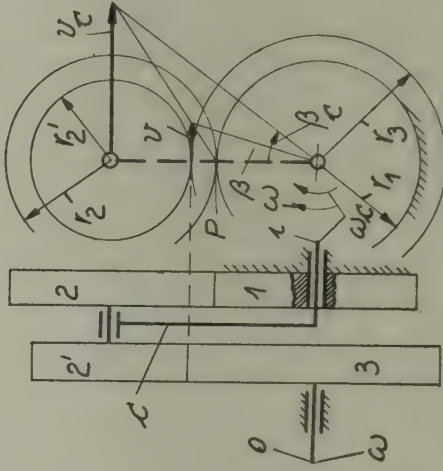
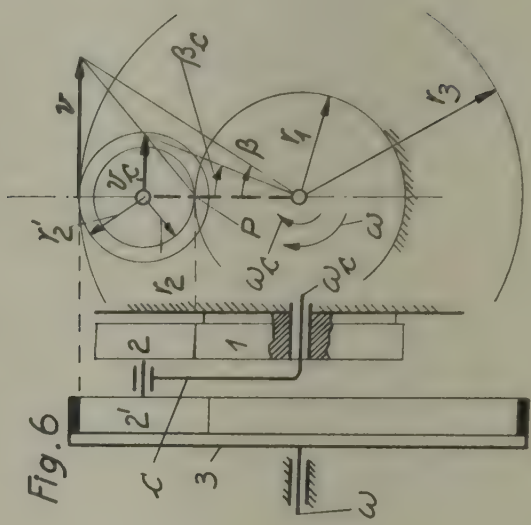


Fig. 6



FIGS. 1-6—SIMPLE EPICYCLIC GEAR TRAINS DEVELOPMENT

Thus with  $t_1=63$  equation (1) gives as the highest ratio of transmission  $R = \frac{\omega_c}{\omega} = -20$  for Fig. 3, while in an ordinary gear of the same size the ratio may be about 6.

In Fig. 4 with the crank or planet carrier  $c$  as the input shaft  $i$ , centre  $C_2$  has the velocity  $v_c = \omega_c b$ , from which we find with the instantaneous centre  $P$  and by parallel lines the revolved vectors  $v'_a$ ,  $v'_b$  and the true vector  $v_2 = a\omega$  with  $\omega$  as output velocity. Vector  $v_2$  points in the same direction of rotation as vector  $v_c$  does, i.e. in- and output shafts rotate in the same direction. Similarity of triangles gives

$$v'_b : v'_c = a : r_2, \text{ therefore} \\ \omega = \omega_c \frac{b}{r_2} = \omega_c \frac{r_2 - r_1}{r_2} = \omega_c \frac{t_2 - t_1}{t_2}$$

equal to equation (1).

#### Four-Gear Trains

Instead of using parallelograms as mentioned before, to get a coaxial gear we may add to the two-gear train another two-gear train and connect the two planet gears into one block. The planet carrier will then also be only one. Thus we get the four-gear train or the compound Epicyclic Gear. As two of the gears may be internal gears, we get thus the variety of gears shown in Figs. 5-13.

The input shaft may be the carrier shaft or a sungear shaft. The fixed sungear shall be always gear 1. Let the angular velocity of the carrier shaft be  $\omega_c$ , and one of the sungear 3 be  $\omega$ . Velocity vectors are denoted by  $v$ , radii of pitch circles by  $r$ . The velocity diagram (Fig. 5) shows that  $v_c = \omega_c (r_1 + r_2) = (r_1 + r_2) \tan \beta_c$ , therefore  $\tan \beta_c$  representing the angular velocity. The line  $g$  from the instantaneous centre  $P$  through the arrow head of  $v$  gives the velocities of all points of the gear block 2, 2' in its centre line. Thus we find vector  $v$  and  $\omega = \frac{v}{r'_3}$ . Similarity of triangles gives

$\frac{v_c}{v} = \frac{r_2}{r_2 - r'_2}$ . Introducing the above values of  $v_c$  and  $v$  we find

$$\omega = \omega_c \left( r - \frac{r_1 r'_2}{r_2 r_3} \right) = \omega_c \left( 1 - \frac{t_1 t'_2}{t_2 t_3} \right) \\ = \omega_c (1 - R_0) \quad \dots \quad (2)$$

with  $t_1, t_2, t'_2, t_3$  as the teeth numbers of the gears.  $R = \frac{\omega_c}{\omega}$  shall be stipulated as ratio of transmission. If in equation (2)  $R_0$  is greater than 1, the velocity  $\omega$  becomes negative, which means opposite direction to  $\omega_c$  (Fig. 5a).

In Fig. 6, we find in the same manner

$$\omega = \omega_c \left( 1 + \frac{t_1 t'_2}{t_2 t_3} \right) = \omega_c (1 + R_0) \quad (3)$$

which indicates that by one internal gear the sign of  $R_0$  becomes changed. In such a gear  $\omega$  and  $\omega_c$  can have the same direction only.

Fig. 7

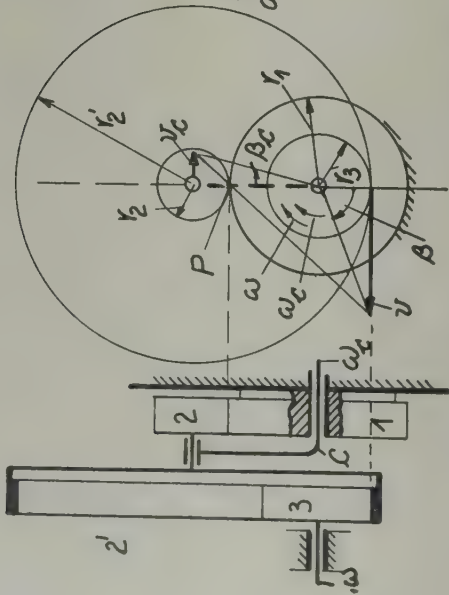


Fig. 8

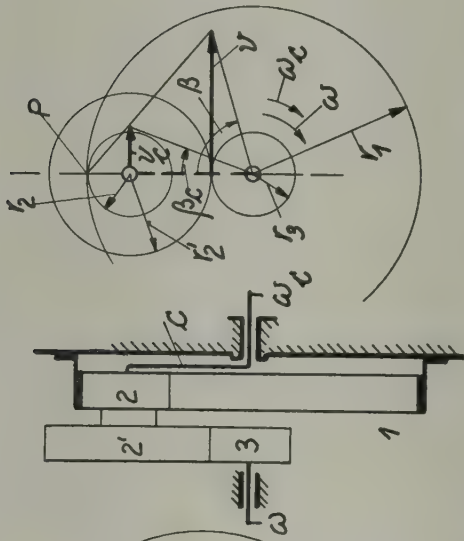


Fig. 9

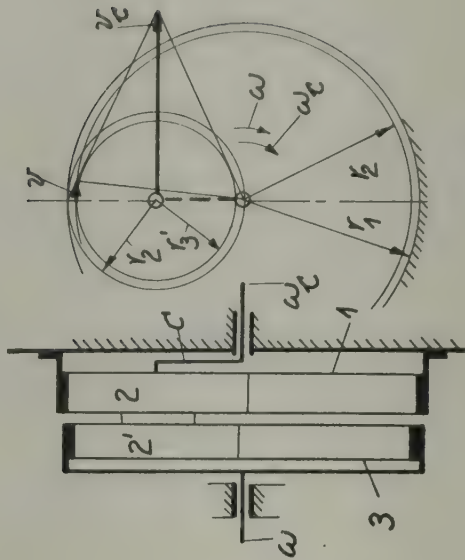


Fig. 10

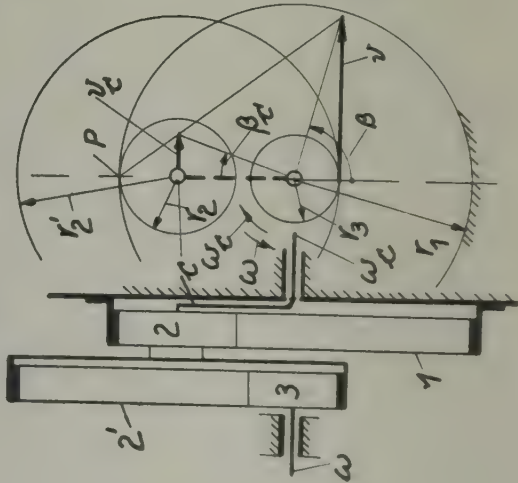


Fig. 11

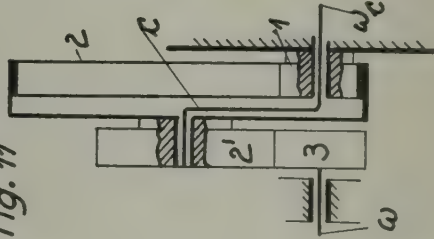


Fig. 12

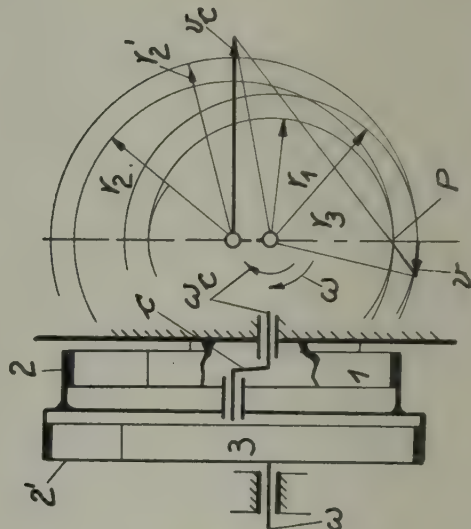
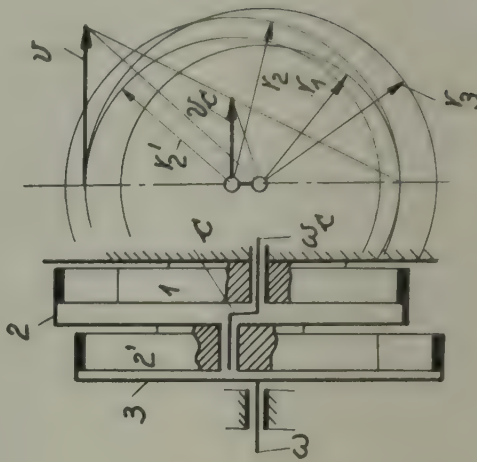


Fig. 13



If we stipulate that for each internal gear, the teeth number or its pitch radius shall have the negative sign, then equation (2) is valid for all epicyclic gears, consisting of four gearwheels and with a sungear fixed.

Figs. 7 and 8, containing one internal gear only, give  $+R_0$  and therefore the same direction of  $\omega_c$  and  $\omega$ .

Gear (Fig. 9), having two internal gears, has  $-R_0$  as in equation (2). Accordingly  $\omega$  may have the same or the opposite direction as  $\omega_c$ .

In Fig. 10 gear 2 is always smaller than gear 1, gear 3 smaller than gear 2'; therefore  $R_0$  is greater than 1 and hence  $\omega$  is always negative. Out- and input shafts can only rotate in opposite directions.

Fig. 11 contains one internal gear. Therefore  $R_0$  is positive and the direction of rotation the same for both shafts.

Fig. 12 with two internal gears has the same direction of rotation for both shafts, if gear 2' is larger than gear 2.

Gear (Fig. 13) has always  $R_0 < 1$  and is therefore to be used only if both shafts shall rotate in the same direction.

### Simplified Four-Gear Trains

Simplifying restricts the free choice of ratio of transmission, but makes the gear simpler, less spacious and cheaper.

A well known simplification is achieved in Fig. 8 by making  $t_2 = t'_2$  as in Fig. 14 with equal pitches thus getting only one planet gear. To get a ratio of transmission, one sungear has to be internal. Therefore equation (2) becomes

$$\omega = \omega_c \left( 1 + \frac{t_1}{t_3} \right) \quad \dots \quad (4)$$

which means the same direction of rotation for both shafts and no high ratio of transmission.

From Fig. 10 we get Fig. 15 with two internal gears; therefore, calculation should be according to equation (2). Sizes of external gears running in internal gears are limited by the addenda and a clearance between the addendum circles to allow a tooth to leave a groove and to pass over to another one. If  $p$  denotes the diametral pitch and if the addendum is taken as  $\frac{1}{p}$  and if the clearance is assumed to have the same value, then it must be

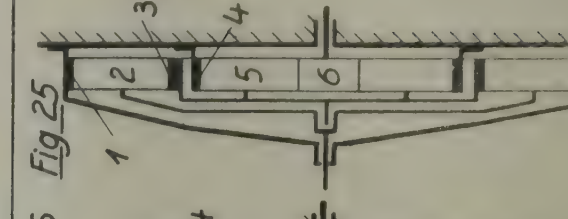
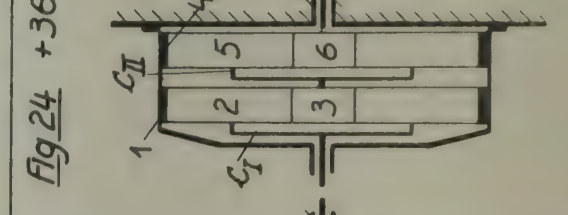
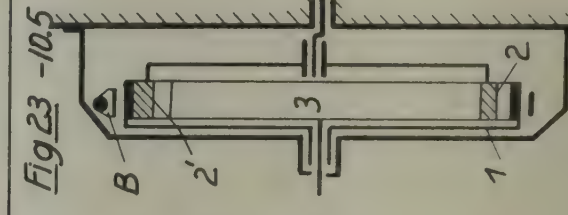
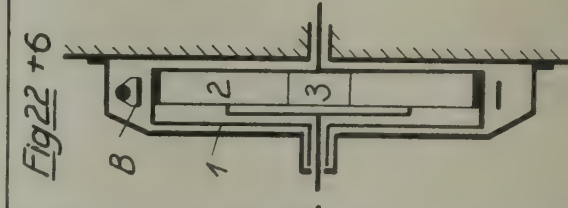
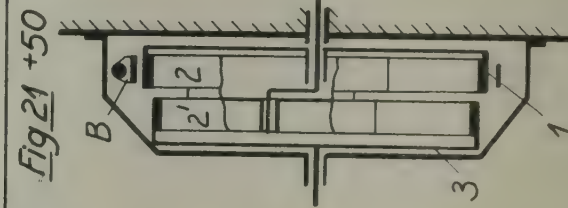
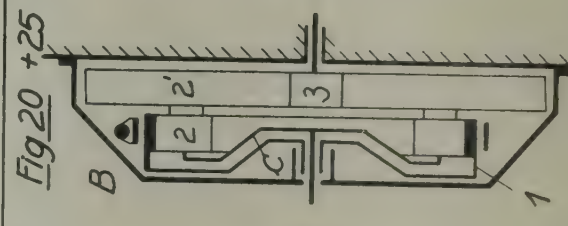
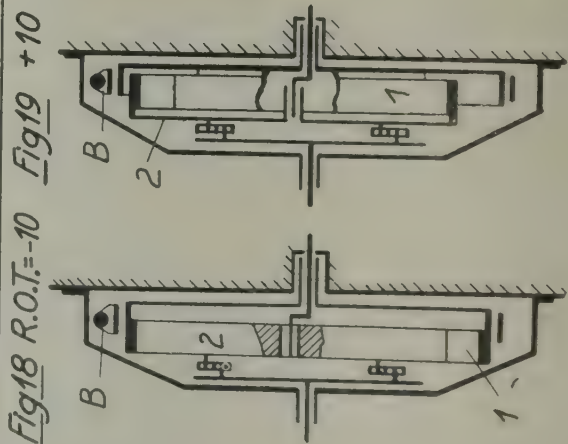
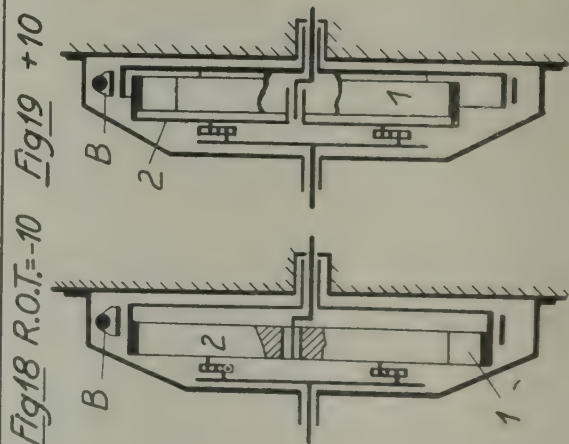
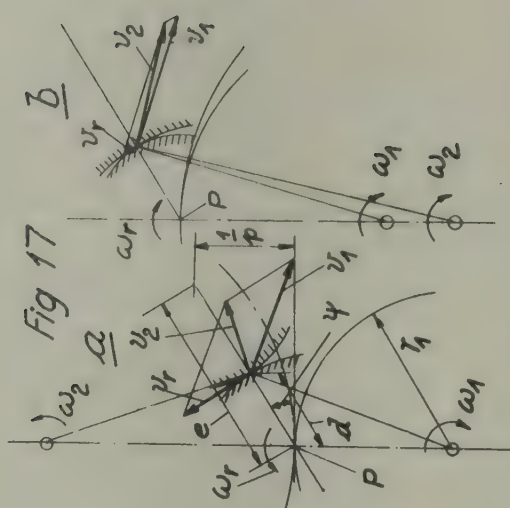
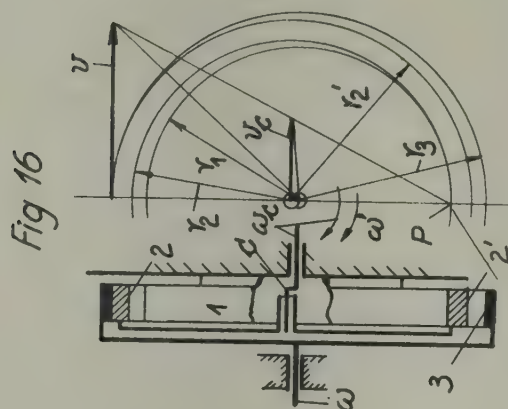
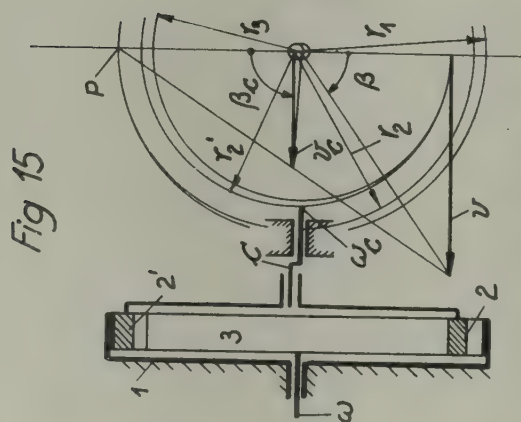
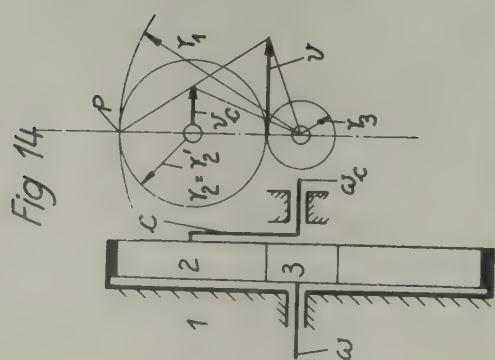
$$2r'_2 = \frac{t'_2}{p} \geq \frac{t_3}{p} + \frac{3}{p} \quad \text{or} \quad t'_2 \geq t_3 + 3$$

If the addendum is  $\frac{1.25}{p}$  and the thickness of the rim is assumed as  $\frac{2}{p}$

$$\text{we have } 2r_2 \geq \frac{t_2}{p} + 2 \times \frac{1.25}{p} + \frac{2}{p} + 2 \times \frac{1.25}{p} \quad \text{or} \quad t_2 \geq t'_2 + 7$$

For gear 1 we find

$$t_1 \geq t_2 + 3$$



FIGS. 14-25—SIMPLIFIED EPICYCLIC GEARS WITH GOOD DEGREES OF EFFICIENCY

We may now write  $t_1 = t'_2 + t_3 = t'_2 + 10 = t_3 + 13$ ,  $t_2 = t_3 + 10$  and get as limit  $R_o = \frac{t_1}{t_2} \times \frac{t'_2}{t_3} = \frac{(t_3 + 13)}{(t_3 + 10)} \frac{(t_3 + 3)}{t_3}$ . If we choose for example  $t_3 = 60$ ,

we find  $\frac{t_1}{t_2} \times \frac{t'_2}{t_3} = \frac{73}{70} \times \frac{63}{60} = 1.095$

Thus we get from equation (2)

$$\omega = \omega_c (1 - 1.095) = -\omega_c \times 0.095 \text{ or } \frac{\omega_c}{\omega} \text{ approximately equals } -10$$

as the highest possible transmission.

From Fig. 13 we develop the simplified gear (Fig. 16) with the limits  $t_3 = t_1 + 13$ ,  $t'_2 = t_1 + 10$ ,  $t_2 = t_1 + 3$  and

$$R_o = \frac{t_1}{t_2} \frac{t'_2}{t_3} = \frac{t_1}{(t_1 + 3)} \frac{(t_1 + 10)}{(t_1 + 13)} = \frac{1}{1.095},$$

if  $t_1 = 60$  is chosen. Equation (2) gives

$$\omega = \omega_c \left( 1 - \frac{1}{1.095} \right) = +\omega_c \times 0.087 \text{ or } \frac{\omega_c}{\omega} = +11.5 \text{ as highest ratio of}$$

transmission.

### Degree of Efficiency

In any gear, when  $P_i$  is the input,  $P$  the output power,  $P_i - P = L$  is the loss of power and  $E = \frac{P}{P_i} = \frac{P_i - L}{P_i} = 1 - \frac{L}{P_i}$  ... (5)

declared as the degree of efficiency. If  $\mu$  denotes the co-efficient of friction,  $F_1$  the force of pressure,  $v_r$  the relative velocity between the grinding surfaces, power loss by friction is  $L = \mu F_p v_r$ . As  $F_p$  is given, loss  $L$  may be lowered by reducing  $\mu$  with the help of good lubrication. To reduce the loss  $L$  further, rolling friction may be created by special gear teeth and rollers as the meshing partners as it is done in the Cyclo Gears in which cycloidal teeth mesh with rollers. In gears with ordinary teeth the relative velocity  $v_r$  will be smaller and the loss  $L$  reduced by using internal gears. Fig. 17a shows that in an external gear the relative angular velocity at the pitch point  $P$  is  $\omega_r = \omega_1 + \omega_2$ , while in an internal gear Fig. 17b it is  $\omega_r = \omega_1 - \omega_2$ , the negative sign occurring for the internal gear wheel. Combining both cases we may write  $\omega_r = \omega_1 \pm \omega_2$ . In the distance  $d$  we have  $v_r = \omega_r d$ , which becomes much smaller in the internal gear than in the external gear. If the addendum is  $\frac{1}{p}$  with  $p$  as diametral pitch and if  $\psi$  represents the pressure angle, the length of the path of contact in a rack is limited by

$$e = \frac{2}{p \sin \psi}.$$

As in epicyclic gears the numbers of teeth are usually high, we may roughly assume 'e' as the true way of contact. The velocity  $v_r$  grows with the distance  $d$ , hence the average value of  $d$  may be taken as  $d = \frac{e}{4} = \frac{1}{2p \sin \psi}$ .

If  $F$  denotes the load tangential to the pitch circle, the force in the line of action is  $F_v = \frac{F}{\cos \psi}$ . The average loss is therefore  $L = \mu F_v v_r = \frac{\mu F \omega_r}{p \ 2 \sin \psi \cos \psi}$ .

With  $\omega_r = \omega_1 \pm \omega_2 = \omega_1 \left(1 \pm \frac{\omega_2}{\omega_1}\right) = \omega_1 \left(1 \pm \frac{t_1}{t_2}\right)$

and  $\omega_1 = \frac{v}{r_1}$  or because  $2r_1 p = t_1$ ,  $\omega_1 = \frac{v 2p}{t_1}$  we get  $\omega_r = 2pv \left(\frac{1}{t_1} \pm \frac{1}{t_2}\right)$ .

Therefore the loss will be  $L = \mu \frac{Fv}{\sin \psi \cos \psi} \left(\frac{1}{t_1} \pm \frac{1}{t_2}\right)$  ... (6)

with the negative sign, if the gear is internal. With  $\psi = 20^\circ$ ,  $\mu = 0.064$  we get

$$L = 0.2 Fv \left(\frac{1}{t_1} \pm \frac{1}{t_2}\right) \quad \dots \quad (6a)$$

Other losses are here neglected, because here gears are compared with each other and these losses are more or less equal in all gears.

In gears with more than two wheels the losses may be summed up as an approximation according to

$$L = L_1 + L_2 = 0.2 P_i \left(\pm \frac{1}{t_1} \pm \frac{1}{t_2} \pm \frac{1}{t_3} \pm \frac{1}{t_4}\right) \quad \dots \quad (7)$$

for a gear with four wheels.  $P_i = Fv$  represents the input power.

In an ordinary gear train with a fixed casing the power  $P_i$  goes over the gear teeth to the output shaft. The degree of efficiency is then according to equation (5) and for instance (6a)

$$E = 1 - 0.2 \left(\frac{1}{t_1} \pm \frac{1}{t_2}\right) \quad \dots \quad (8)$$

In an epicyclic gear (Fig. 5) the velocity of the fixed gear 1 against the carrier is  $v = -\omega_c r$  with  $+\omega_c$  as angular velocity of the carrier and  $r_1$  as pitch radius of gear 1. The load at the pitch point is with  $M_i$  as input torque.

$$F = M_i \frac{r'_2}{(r_1 + r_2)(r_2 - r'_2)} = M_i \frac{r'_2}{r_2 r_3 - r_1 r'_2}$$

In equation (6a) or (7) we have  $Fv = -M_i \omega_c \frac{r_1 r'_2}{r_2 r_3 - r_1 r'_2}$  with  $M_i \omega_c$

as input power at the carrier. From equation (2) we find

$$\frac{r_1 r'_2}{r_2 r_3 - r_1 r'_2} = \frac{1}{\frac{1}{R_0} - 1} = \frac{R_0}{1 - R_0} = \frac{\omega_c}{\omega} R_0$$

$$= \frac{\omega_c}{\omega} \left(1 - \frac{\omega}{\omega_c}\right) = \frac{\omega_c}{\omega} - 1. \quad \text{Equation (5) becomes now}$$

$$E = 1 - \frac{L}{P_i} = 1 - 0.2 \left(\frac{-\omega_c}{\omega} + 1\right) \left(\pm \frac{1}{t_1} \pm \frac{1}{t_2} \pm \frac{1}{t_3} \pm \frac{1}{t_4}\right) = 1 - |T|L \quad \dots \quad (9)$$

TABLE OF EFFICIENCY OF GEARS

Gear	For speed reduction, driver	Teeth numbers					R	T	L	E	TN
		$t_1$	$t_2$	$t'_2$	$t_3$						
Fixed Carrier	Gear 1	+ 12	+ 120	—	—		— 10	1	0.01833	0.9817	132
do.	Gear 1	+ 12	— 120	—	—		+ 10	1	0.01500	0.9850	132
Fig. 3	Carrier	— 66	+ 60	—	—		— 10	11	0.00030	0.9967	126
do.	do.	— 33	+ 30	—	—		— 10	11	0.00060	0.9934	66
Fig. 4	Carrier	+ 54	— 60	—	—		+ 10	9	0.00037	0.9967	114
do.	do.	+ 27	— 30	—	—		+ 10	9	0.00074	0.9933	57
Fixed Carrier	Gear 1	+ 12	+ 120	+ 12	+ 120		+ 100	1	0.03667	0.9633	264
do.	do.	+ 12	— 120	+ 12	+ 120		— 100	1	0.03333	0.9667	264
do.	do.	+ 12	— 120	+ 12	— 120		+ 100	1	0.03000	0.9700	264
Fig. 5	Carrier	+ 49	+ 49	+ 48	+ 50		+ 25	24	0.013	0.688	196
Carrier	Carrier	+ 51	+ 51	+ 52	+ 50		— 25	26	0.0157	0.5918	204
Fig. 8	Gear 3	— 60	+ 10	+ 40	+ 10		+ 1	24	0.0383	0.9632	120
Fig. 9	Carrier	— 50	+ 46	+ 45	— 49		+ 25	25	0.00070	0.6118	190
							+ 555.56	554.56			
Fig. 10	Gear 3	— 33	+ 25	+ 24	— 32		+ 100	99	0.00402	0.6020	114
		— 99	+ 75	+ 72	— 96		+ 100	99	0.00134	0.8673	342
Fig. 12	Carrier	— 63	+ 45	+ 42	— 60		+ 50	49	0.0027	0.8677	210
		— 48	+ 32	+ 34	— 50		— 50	51	0.0040	0.7960	164
Fig. 14	Gear 3	— 108	+ 36	— 84	+ 12		+ 1	21	0.0201	0.9789	240
							— 20	20			
Fig. 15	Carrier	+ 42	— 60	— 63	+ 45		+ 50	49	0.0027	0.8677	210
Fig. 16	Carrier	+ 42	— 70	— 68	+ 40		— 50	51	0.0040	0.7960	220
		— 60	+ 24	—	+ 12		+ 1	5	0.03	0.9750	96
							+ 6	6			
Fig. 15	Carrier	— 73	+ 70	— 63	+ 60		— 10.5	11.5	0.00028	0.9678	266
Fig. 16	Carrier	+ 60	— 63	+ 70	— 73		+ 11.5	10.5	0.00028	0.9943	266

Angle of action  $\Psi = 20^\circ$ . Co-efficient of friction  $\mu = 0.064$ . Transmission factor  $T = \left| 1 - \frac{\omega_c}{\omega} \right|$ , loss factor  $L = 0.2 \sum \pm \frac{1}{t}$  (+t teeth number for external, -t for internal gear)  $R = \omega_c / \omega$  = Ratio of transmission in epicyclic gears,  $R = \omega_1 / \omega_N = t_s / t_1$  in fixed gears. Degree of Efficiency  $E = 1 - |T|L$ . TN = Total teeth number.

where the 'transmission factor'  $|T| = \frac{\omega_c}{\omega} + 1$  is to be used with its positive or absolute value only, because  $T$  represents a ratio of energies.  $L = 0.2 \sum \pm \frac{1}{t}$  is the 'loss factor'. Equation (8) is equally valid if gear 3 is the driving member. With  $\omega_c = 0$ , i.e., if the carrier is fixed, equation (9) becomes equation (8).

It is important to note that in all equations the loss factor will be lowered by higher numbers of teeth.

### Limits for Ratios of Transmission, Degrees of Efficiency

For a high ratio of transmission the value  $R_0$  in equation (2) must be very nearly 1. At the same time, if all gears have the same pitch, the centre distances of two meshing gears must be the same, i.e.,

$$|\pm t_1 \pm t_2| = |\pm t'_2 \pm t_3| \quad \dots \quad (10)$$

where the negative signs are for internal gears.

In Fig. 5 we get with  $R_0 = \frac{99}{100} \times \frac{99}{100} = 0.9801$  the ratio  $R = \frac{\omega_c}{\omega} = +50.25$ .

For Fig. 8, equation (10) becomes  $t_1 - t_2 = t'_2 + t_3$ . If with gear 3 as driver we choose  $\frac{\omega_c}{\omega} = \frac{1}{R} = +25$ , the positive sign meaning the same direction of both angular velocities, we find from equation (3),  $R_0 = 24 = 4 \times 6$ . Assuming  $t_2 = t_3 = 10$  as the smallest possible tooth number we have  $R_0 = \frac{40}{10} \times \frac{60}{10}$ . These numbers fulfil in the sequence  $60 - 10 = 50 = 40 + 10$  the equation (10). Accordingly it has to be made  $t_1 = 60$ ,  $t'_2 = 40$ . The transmission factor becomes  $T = \left| 1 - \frac{1}{25} \right| = \frac{24}{25}$ , the loss factor  $L = 0.2 \left( \frac{-1}{60} + \frac{1}{10} + \frac{1}{40} + \frac{1}{10} \right) = 0.0416$ , hence the degree of efficiency  $E = 0.9600$ .

The gear (Fig. 9) has the limits  $t_1 = t_2 + 3$ ,  $t_3 = t'_2 + 3$ . Equation (10) becomes  $t_1 - t_2 = t_3 - t'_2$ . If we choose  $t_1 = 50$ ,  $t_3 = 49$ ,  $t_2 = 46$ ,  $t'_2 = 45$  and equation (10) is satisfied by the difference,  $50 - 46 = 49 - 45 = 4$ . From equation (2) we find  $R = \frac{\omega_c}{\omega} = 555.56$ . We get  $T = |1 - 555.56| = 554.56$  and  $L = 0.2 \left( -\frac{1}{50} + \frac{1}{46} + \frac{1}{45} - \frac{1}{49} \right) = 0.0007$ , therefore the degree of efficiency  $E = 0.6118$ . If the teeth numbers are chosen three times as high, the loss factor will become one-third and the degree of efficiency  $E = 0.8636$ . In these gears the carrier is the driver.

Fig. 10 is useful with gear 3 as driver and opposite directions of shaft revolutions. For  $\frac{\omega_c}{\omega} = -20 = -\frac{1}{R}$ , we find from equation (20),  $R_0 = R_1 = \frac{3}{1} \times \frac{1}{7}$  or for equal teeth differences,  $R_0 = \frac{108}{36} \times \frac{84}{12} = \frac{t_1}{t_2} \times \frac{t'_2}{t_3}$ . We get  $T = \frac{21}{20}$ ,  $L = 0.0201$ ,  $E = 0.9789$ .

Gear (Fig. 11) is not suited for a remarkable ratio of transmission.

In Fig. 12 with the carrier as driver it may be given  $R = +50$ .

Equation (2) gives  $R_o = \frac{49}{50} \times \frac{7 \times 7}{5 \times 5 \times 2}$ . Since  $t_2 > t_1$ ,  $t'_2 > t_3$ , we write

$$R_o = \frac{t_1}{t_2} \times \frac{t'_2}{t_3} = \frac{7}{10} \times \frac{7}{5}.$$

For equal differences  $t_2 - t_1 = t'_2 - t_3$  we multiply the numerator and denominator of the left fraction with the difference of numerator and denominator of the right fraction and *vice versa*, thus getting

$$R_o = \frac{14}{20} \times \frac{21}{15}.$$

For higher teeth numbers and a better degree of efficiency we expand the fractions by the same factor for instance to  $R_o = \frac{42}{60} \times \frac{63}{45}$ . We find  $T = 49$ ,  $L_f = 0.0027$ ,  $E = 0.8677$ .

Fig. 13 is not suitable for a high ratio of transmission.

### Gears for Disengagement

For a start without load gear 1 is to be fixed by a friction clutch which may be a band brake B for high torques as indicated in Figs. 18-22, and thus may be released for starting the engine. To avoid load on the shafts by the brake a double block brake with balance of block pressure can be used. Table 1 gives some details of short and simplified gears for disengagement.

TABLE 1—DETAILS OF SHORT AND SIMPLIFIED GEARS FOR DISENGAGEMENT

	In-/output velocity	Efficiency
Fig. 18 from Fig. 3	-10	0.9967
Fig. 19 from Fig. 4	+10	0.9967
Fig. 20 from Fig. 8	+25	0.9600
Fig. 21 from Fig. 9	+50	0.8677
Fig. 22 from Fig. 14	+6	0.9750
Fig. 22 from Fig. 15	-10.5	0.9678

### Simple Epicyclic Gears in Series

A still comparatively simple gear with a higher ratio of transmission is achieved by using two gears (Fig. 14) and connecting the carrier of one gear with the sun gear of the other one as it is done in Fig. 24. If  $\omega_i$  denotes the angular velocity of the driving gear 6,  $\omega_c$  the velocity of the carrier  $C_{II}$  and  $\omega$  the output velocity of the carrier  $C_I$ , we find from equation (4)

$$\omega_i = \omega_c \left( 1 + \frac{t_4}{t_6} \right) \text{ and } \omega_c = \omega \left( 1 + \frac{t_1}{t_3} \right), \text{ therefore}$$

$$\omega_i = \omega \left( 1 + \frac{t_1}{t_3} \right) \left( 1 + \frac{t_4}{t_6} \right). \quad \dots \quad (II)$$

Fig. 26

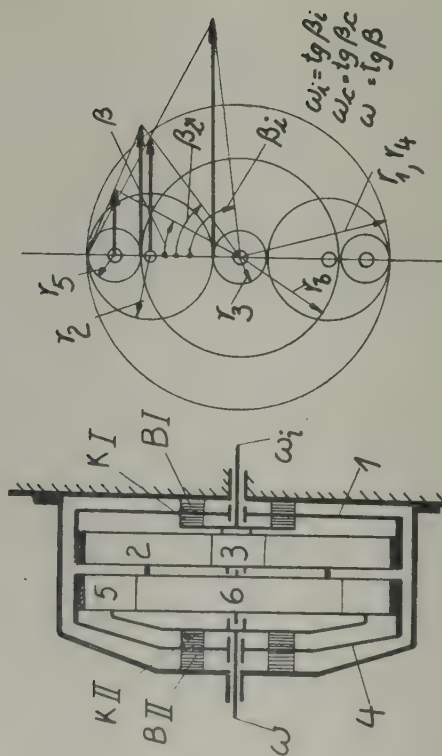


Fig. 27

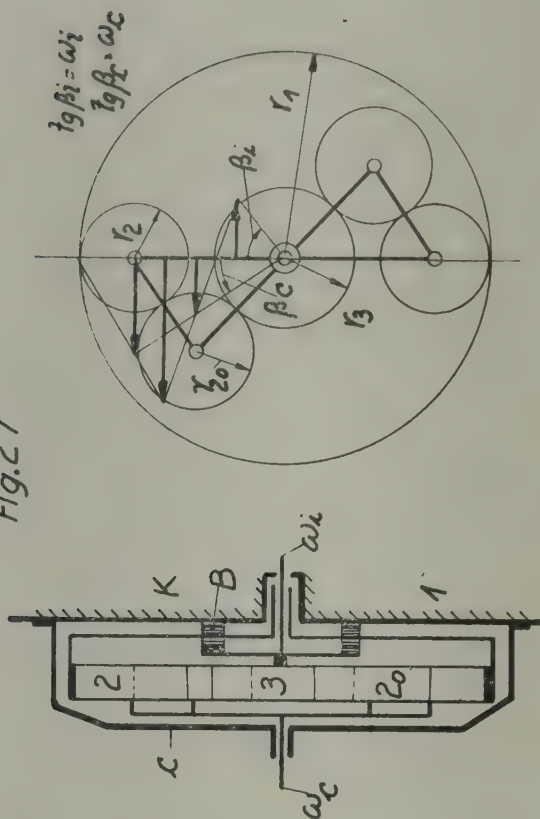
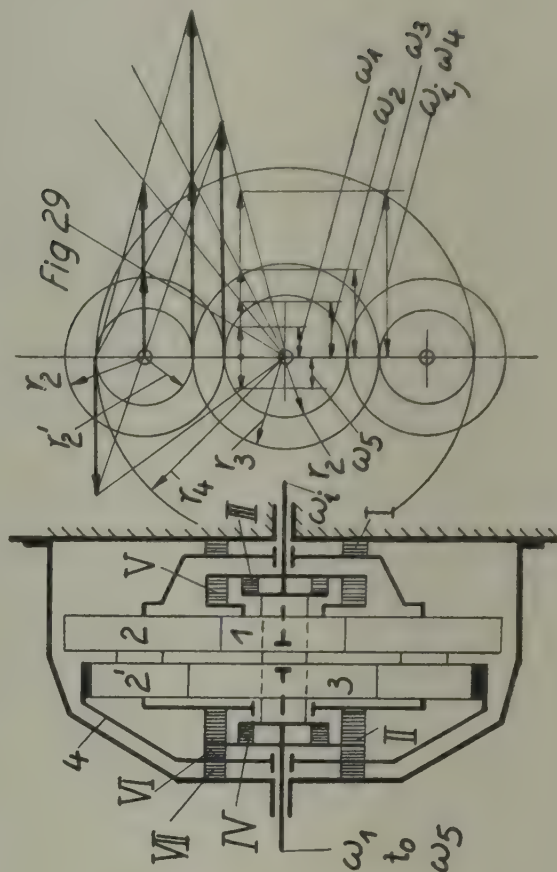
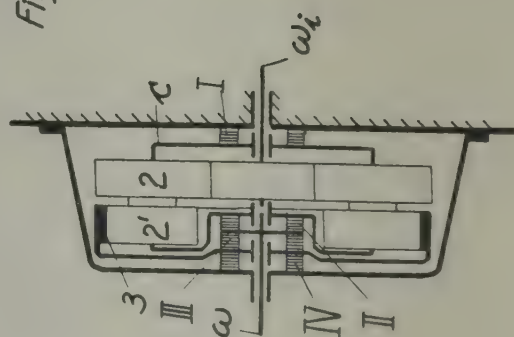


Fig. 28



FIGS. 26-29—SIMPLE EPICYCLIC CHANGE SPEED GEARS

If we choose  $t_1=60$ ,  $t_3=12$ ,  $t_4=60$ ,  $t_6=12$ , we get in-output velocity  $\omega_i/\omega = +36$  and a degree of efficiency  $E=E_1 \times E_2=0.975^2=0.9506$ .

The same gear can be built with a shorter length as shown in Fig. 25 if  $t_3 \geq t_4 + 7$ . If we assume  $t_4=60$ ,  $t_6=12$ ,  $t_1=102$ ,  $t_3=68$ , when  $t_5=24$ ,  $t_2=17$ , equation (11) gives  $\omega_i/\omega = 15$ .

### Simple Change Speed Gears

Gear (Fig. 24) is easily carried out as a change-speed gear (Fig. 26) by making gear 1 and 4 movable and fixing them with brakes BI, BII. If we add two clutches KI, KII to prevent relative motion within the gears, we are able to achieve four different speeds by opening or closing the brakes and clutches according to the following table (O=open, C=closed) for gear (Fig. 26) with  $\omega_i$  as input,  $\omega_1$  to  $\omega_4$  as output velocities.

$\frac{\omega_i}{\omega}$	BI	BII	KI	KII
15	C	C	O	O
6	C	O	O	C
2.5	O	C	C	O
1	O	O	C	C

If the ratio of transmission of the first gear is  $R_1 = \frac{\omega_i}{\omega_c} = 6$ , of the second gear  $R_{II} = \frac{\omega_c}{\omega} = 2.5$ , the highest ratio will be  $R = R_1 \times R_{II} = 15$  and for the direct drive  $R_d = 1$  as stated in the above table. From equation (3) we find  $R_{01} = 5 = \frac{t_1}{t_3}$ ,  $R_{0II} = 1.5 = \frac{t_4}{t_6}$ .

To get a change-speed gear with only a direct drive and one slower reverse drive, we introduce in Fig. 21 an idler 2<sub>0</sub> according to Fig. 27 and besides the brake B a clutch K to create the direct drive, when B is open. Closing of B and opening of K gives the reverse drive, opening of B and K disengagement. Equation (2) gives for the reverse speed

$$\frac{\omega_i}{\omega_c} = 1 - \frac{t_1}{t_3} \quad \dots \quad (12)$$

Epicyclic gears can be combined in many other ways. Combination in series allows output velocities in steps of a geometrical sequence. The highest number of speeds with the lowest number of wheels but with restriction of choice of speeds is achieved with the method of changing connections, i.e. by means of hydraulic, pneumatic or magnetic clutches to fix a sungear or the planet carrier either to the input/output shaft or to the casing. Fig. 28 shows a gear in which all these possibilities are not used, because it is assumed that only a direct drive  $\omega_3 = \omega_1$ , one lower speed  $\omega_2$  in the direction and one reverse speed  $\omega_1$  are needed. Clutch I connects the carrier  $c$  with the casing clutch II with the output shaft. Clutch III connects the gear 3

with the output shaft, clutch IV with the casing. The table for operation of the clutches is the following.

Speed	$\omega_i/\omega$	Clutch			
		I	II	III	IV
$\omega_1$	-4	C	O	C	O
$\omega_2$	+5	O	C	O	C
$\omega_3$	+1	O	C	C	O

It is  $\omega_1 = -\omega_i \frac{r_1}{r_2} \frac{r'_2}{r_3} = -\frac{\omega_i}{R}$  and  $\omega_2 = \omega_1 \frac{I}{I+R_o}$ . If we assume  $R_o=4$  we get  $\omega_1 = -\frac{1}{4} \omega_i$ ,  $\omega_2 = +\frac{1}{5} \omega_i$ .

Gear (Fig. 29) contains one more sungear than Fig. 28, but can give seven speeds with the maximum number of nine clutches. In Fig. 29 only seven clutches are installed to obtain five speeds, one of them reversed. Clutches I and II connect the carrier C with the casing or with the output shaft. Sungear 3 is fixed to a long hub which can be connected by clutch III with the input shaft  $i$ , by clutch IV with the output shaft O. Clutch V permits a connection of gear 1 with the input shaft  $i$ . With clutches VI and VII sungear 4 can be fixed to the output shaft O or to the casing. To obtain the speed  $\omega$  from the input speed  $\omega_i$  the clutches have to be operated according to the following table (C=closed, O=open), of which the speeds are calculated below.

Speed	$\omega_i/\omega$	Clutch						
		I	II	III	IV	V	VI	VII
$\omega_1$	6	O	C	O	O	C	O	C
$\omega_2$	3	O	C	C	O	O	O	C
$\omega_3$	2	O	O	O	C	C	O	C
$\omega_4$	1	O	O	C	C	O	O	O
$\omega_5$	-5	C	O	O	O	C	C	O

$$\left[ \omega_1 = \omega_i \frac{I}{I + \frac{t_2}{t_1} \times \frac{t_4}{t'_2}}, \quad \omega_2 = \omega_i \frac{I}{2 \left( I + \frac{t'_2}{t_3} \right)} = \frac{1}{2} \omega_i \left( I - \frac{I}{\frac{t_4}{t_2} - I} \right) \right]$$

$$\left[ \omega_3 = \omega_i \frac{\frac{t_4}{t_3} - I}{\frac{t_4}{t_1} - I} = \omega_i \frac{I}{I + \frac{t_4}{2t_1} \left( \frac{t_2}{t'_2} - I \right)}, \quad \omega_4 = \omega_i, \quad \omega_5 = -\omega_i \frac{t_1}{t_2} \frac{t'_2}{t_4} \right]$$

Assuming  $\omega_i : \omega_1 = 6$ ,  $\omega_i : \omega_2 = 3$ , we find from  $\omega_1$  the ratio  $\frac{t_2}{t_1} \times \frac{t_4}{t'_2} = 5$ .

Therefore  $\omega_1 : \omega_5 = -5$ , and from  $\omega_2$  the ratio  $t_4 : t'_2 = 4$ , therefore  $t_2 : t_1 = 5 : 4$ . If we assume  $t'_2 = 18$  as the smallest pinion, we get  $t_4 = 72$ . Since for equal centre distances  $t_4 - t'_2 = t_1 + t_2 = 54$ , we get  $t_1 = 54 \times 4/9 = 24$ ,  $t_2 = 30$  and from  $t_4 - 2t'_2 = t_3$ , finally  $t_3 = 36$ . We find now  $\omega_1 : \omega_3 = 2$  as stated in the above Table.



# Manufacture of Fuel Injection Equipment

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## Introduction

Fuel injection equipment is often called the heart of the compression ignition engine for which it is essential that an accurately metered quantity of fuel be injected in the form of a fully atomized spray at the proper moment when the compression stroke is near its end. Basically, the fuel injection equipment consists of one or more fuel filters, a fuel pump with its actuating cam, and an atomiser or injector to create the necessary spray. Plunger type of pumps are used almost exclusively apparently due to their reliability in operation under the high pressures involved.

On the average a diesel engine consumes about half a pound of fuel per hour per brake horse power output. At this rate, a unit brake horse power four-stroke cycle engine running at 1,000 r.p.m. requires one-sixty-thousandth pound of fuel per cycle. To deliver such small quantities of fuel necessitates nozzles with very small holes and pumps with very small dimensions ; for instance, plunger diameter is often as small as 5 mm. and is seldom more than 20 mm. The pressure to which the fuel has to be compressed to be able to spray it in the compressed air in the cylinder is high, often as high as 4,000 lb./sq. in. Such high pressures impose a number of serious problems, such as the problem of leakage past the plunger. To keep the leakage within tolerable limits, the clearance between the plunger and the barrel has to be of the order of 0.0001 in. The equipment is further complicated by the fact that the pump should not only deliver very small amounts of fuel at considerably high pressures, it should also be able to regulate closely the extremely small amounts of fuel being pumped, since different amounts are required for different loads. The design and manufacture of fuel injection equipment therefore presents a number of difficult problems.

There are several different types of fuel pumps and injectors in use ; but the most common types in India, especially for the locally manufactured engines, are the pumps and injectors manufactured by the Robert Bosch Company and the Bryce Company.

In the Bosch pump, a plunger is driven up and down inside a barrel by means of a cam and spring. The inlet and spill ports made in the barrel are controlled by the motion of the plunger and the delivery is effected by means of a spring loaded discharge valve fitted at the upper end of the barrel. To enable the pump to vary the quantity of fuel delivered, the plunger has a vertical channel extending from its top edge to an annular slot which has a circular lower end and a helical upper edge. By turning the plunger, the helical edge of the slot cuts off the delivery by communicating to the spill port. The plunger is turned by operating a toothed sleeve, which meshes

with the teeth of a rack connected to the governor control and the manual control. Two slots on the toothed sleeve engage the lugs of the plunger at its lower end.

A Bosch nozzle consists of two parts, the nozzle valve and the nozzle body. The nozzle valve takes the form of a circumferentially grooved barrel which is ground to fit the nozzle body. At one end of the nozzle valve a stalk is provided, while at the other end of the nozzle there are small tunnels bored vertically in the nozzle body terminating in an annular reservoir just above the valve seat. The nozzle valve is held down on its seat by means of a spring acting through a spindle, and the compression placed upon the spring can be adjusted by a compression screw.

The total proposed annual installed capacity for the manufacture of diesel engines in India is about 40,000. Judging from the recent trends, the consumption will be perhaps much higher at the time all the proposed capacity has been installed. The target provisionally fixed by the Planning Commission in the Five Year Plan of increased production of diesel engines is 55,620 by 1955-56.

Although attempts are being made to start local manufacture of injection equipment, all the equipment required at present has to be imported. The injection equipment has to be imported not only for the new engines, locally manufactured as well as imported, but also for the replacement of parts required for a large number of older engines already in use. The nozzles of the fuel pumps generally need replacement after every one or two years. The total number of diesel engines imported and locally manufactured during the past ten years is about 1,25,000. The present demand of nozzles for replacement may be estimated at 75,000. This is in addition to the injection equipment required for 40,000 new engines. Considering further the general trend of increasing use of diesel engines, it would seem advisable to aim at a production of complete injection equipment for 55,000 engines and additional nozzles for 1,00,000 engines annually.

The cost of the injection equipment varies with the size of the engine. For the popular sizes in demand in India, the cost of the injection equipment has been roughly estimated to be 9 per cent of the cost of the engine. At this rate, the cost of the injection equipment in demand at present both for the new and the old engines amounts to about eleven million rupees annually. An annual production of injection equipment for 55,000 engines, and additional nozzles for 1,00,000 engines, as suggested above, would be worth about 14 million rupees.

### **Materials and Manufacturing Methods**

The manufacture of injection equipment is essentially a precision job requiring special materials as well as special manufacturing facilities.

The pump barrel and plunger are generally made of forged steel, and are lapped so that the clearance between them is extremely small. Oil tightness is dependent largely on the fit between the plunger and barrel, and the sealing effects of viscous fuel. The plunger must be comparatively long

as it helps to reduce the leakage to a minimum. The inlet ports, which are made in the wall of the pump barrel, must be drilled accurately. The Bosch plunger is provided with a circumferential slot having circular and helical edges. These must be machined to a high degree of accuracy as on them depends the correct metering of fuel. The sleeve which controls the fuel delivery is provided with spur teeth at its upper end and two slots at the lower end to guide the plunger arms. It requires special skill to make it. The oil seatings must be very hard and accurately ground to corresponding shapes, since they have to be exposed to high oil pressures. The seating is generally made very small so as to make the valve leak-proof.

Nozzle body and nozzle valve are generally made of good steel. Nozzle valve after machining accurately has to be hardened, ground and lapped into the nozzle body to the finest possible limits within which it works freely. The abutting faces of the nozzle body and nozzle holder are accurately ground. The nozzle body is made of high grade steel, accurately machined and hardened. At its lower end, it has very small diameter holes requiring skill and patience. In the nozzle holder is drilled accurately an axial hole within which a spindle made of hard steel and machined carefully will reciprocate freely. The other components of the nozzle holder also require special attention.

Should the injection equipment be manufactured in a centralised place in India or should it be manufactured individually by the various manufacturers of diesel engines—is a question to which careful thought should be given. There are some who think that the final product has the best quality when produced 'under one roof', while others maintain that superior results may be achieved by specialisation, particularly in the case of components which call for a special type of skilled labour. In U.S.A. and U.K. the injection equipment was manufactured in the early days individually by the various engine manufacturers ; but, it appears that gradually many of the diesel engine manufacturers found it worthwhile to discontinue the manufacture of their own injection equipment which required highly skilled labour and machinery of high precision. At present, there are few firms which manufacture their own injection equipment—almost all of them buy such equipment from a very small number of concerns which specialise in producing them.

### **Factory Equipment and Labour**

The fineness required in the manufacture of injection equipment requires machinery of high precision. Precision lathes and drilling machines are of utmost importance. It is not uncommon to have to drill holes of the order of 0.008 in. in spray nozzle. The lapping of the plunger and the barrels is often done with a clearance not to exceed 0.0001 in. The factory equipment should therefore be selected in such a manner that accuracies and special processes can be achieved. The equipment in general would consist of high precision lathes, shapers, and drilling, milling and grinding machines along with other mechanical tools of moderate precision. The layout of the equipment should be such as to co-ordinate and facilitate the handling of materials in the best and most efficient manner.

# Mechanical Wear and Internal Combustion Engines

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Wear or abrasion occurs in various forms in all objects, in structural parts, tools, machines, cylinders and piston rings when two bodies mate and move against each other under pressure. Fundamentally as many forms of wear are possible as the types of metallic contacts. Interesting to the industry are two types of wear: one due to rolling friction and the other due to gliding friction. In the following, wear due to gliding friction will be mainly dealt with in relation to the specific problem of cylinder and piston ring wear.

Earliest hypothesis of friction is due to Amontons<sup>1</sup> who stated that the force of friction was due to surface asperity and that it was equal to the tangential force necessary to move the irregularities of one surface past those on the other when a normal force due to a load is acting on them. Coloumb<sup>2</sup> went a step further to say that the resistance to sliding was due to mechanical interlocking of the surface asperities and the force of friction was that required to tear them apart. Both fail, however, to account for the energy required for the friction and how it is expended.

More recent theories by Ewing,<sup>3</sup> Hardy,<sup>4</sup> Tomlinson,<sup>5</sup> Finch,<sup>6</sup> Bowden<sup>7,8,9,10</sup> and others speak of friction in terms of molecular attraction. They suggested that friction is due to the interaction of molecular forces when two bodies are in sliding contact and their surface molecules pass each other. This surface cohesive force is the dominant factor of frictional wear. Bowden's work shows the importance of molecular cohesion and the welding of points in sliding contact, which is a possible means of metal loss from surfaces during wear. In their work Bowden and others have shown that a load applied on sliding metals is carried by a small fraction of the surface area and that there may be instantaneous temperature rise between the sliding metal contacts reaching the melting point of the metals concerned or that they slide past each other with a stick-slip movement. Additional evidence shows the finest of surfaces, be they ground finished, lapped, or polished, are rough molecularly and when two surfaces are in sliding contact and under load, discrete localities are loaded beyond their yield point though the applied load is sufficient only to deform the metal elastically. This plastic yielding causes molecular adhesion and a cold weld is formed. With subsequent sliding these are torn apart and chunks of metal are removed. Bowden and Young<sup>11</sup> working on metallic surfaces point out that these welds are weak when the metallic surfaces are exposed. This is due to the oxide layer formation or the lubricant forming a hydrodynamic layer or the lack of affinity for molecular attraction of the surfaces. In such cases there is little surface damage. Clean

metals on the other hand show greater attraction and molecular bonding. This is increased with further work and the surface damage consequently is higher.

Another criterion is proper selection of mating materials. Soft low melting point alloys are most susceptible to surface welding, but they suffer little damage because of the ease with which they break a welded bond by local softening. When mated with a hard metal the soft metal is found to stick to the harder metal in a very minute scale. Quantitative estimation of this by radio tracer techniques have been attempted to prove the adhesion theory.<sup>12</sup>

Strain hardening is another cause of metal failure when in intimate contact and in that the elastic limit of strain plays an important role. The worked layer seems to be in a different state of stress than the bulk below it, which is enough to crack and remove chunks of metal with subsequent work.

Another contention about metallic wear is that the products of basic wear or the wear products are further responsible for destructive wear.<sup>13</sup>

From the work so far published we can specifically say that there is no empirical law for wear nor does it obey any dimensional analysis. It does not exist apart from service conditions. Regarding types of wear Burwell<sup>14</sup> includes such terms as cutting wear, abrasion, scuffing or galling and pitting fatigue. Cutting wear takes place when a rough surface is mated with a smooth surface (specially when the former is harder) and metal removed from the smoother surface is quite enormous. Surface finish is the determining factor in such a type of wear. Abrasive wear—which is the chief cause of metal deterioration in cylinder and piston rings—is a type of cutting wear with some foreign matter like grit or metallic particles. The determining factors are the hardness of the abrasive particles, their size and degree of angularity. Their effectiveness also depends on service condition like whether the surfaces are mating dry or wet. Sometimes the grit embeds in surfaces and the relative embeddability governs the amount of wear.

Galling or scuffing occurs from a momentary failure of oil film causing momentary temperature rise and plastic yielding of the skin surface. Micro welding and subsequent breaking up of weld may also take place causing material damage.

*Pitting*—It is due to compressive fatigue strength of the material being exceeded when two surfaces are in rolling contact under load. When the stresses set up exceed the endurance limit, crack starts up on the surface and the material flakes out.

*Corrosive wear*—There are various causes for this type of wear, the chief being (1) the inherent property of metal surfaces to get oxidised, (2) the effect of lubricant, (3) the effect of combustion products, in forming corrosive acids.

### Factors Affecting Wear

Burwell<sup>15</sup> summarizes the factors entering into wear in the following three categories: (1) primary operating conditions (2) intermediate factors (3) manifestation of wear.

The first factor deals with conditions of operation like speed, load and composition, and lastly the nature of rubbing material itself.

The second factor includes properties of materials like hardness, toughness, etc., the initial state of surfaces, presence and effect of foreign particles, atmospheric effect and lubrication and its effect.

The third factor deals with wear and corrosion products.

In general the following factors have to be considered before undertaking a wear experiment:

(A) *Relating to the metal*: (1) Metallurgical origin, (2) casting conditions—effect of temperature, (3) chemical composition, (4) shaping conditions—finishing temperature, direction of rolling, etc., (5) presence of impurities, (6) grain size and (7) constitution and heat-treatment.

(B) *Conditions in Service*: (1) Surface condition, (2) material with which the metal is in contact, (3) contact pressure, (4) abrasion speed duration of the abrasion period, (5) dry or wet condition in service and type of lubricant used, (6) operating temperature, (7) presence of abrasive particles, (8) stress corrosion effect and (9) environmental condition.

### Cylinder and Piston Ring Wear

Wear of these parts has been given more consideration perhaps than any other in combustion engines since their wear limits the useful life of the engine. When we say that the engine life is over it does not mean that it has lost much of its substance. According to Shidle,<sup>16</sup> when a 5 ton truck wears out it might have lost only 5 lb.

Many conflicting hypotheses have been offered regarding wear of cylinder liners and rings. It will be of considerable help to review the various fundamental causes. It is confirmed that wear in cylinder and rings is due to abrasion, galling and corrosion. We shall deal with firstly the theoretical aspects and later the metallurgical considerations.

The cylinder liner or sleeve is a cylinder bore located rigidly in the cylinder block and can be removed for replacement. The piston is a means of transmission of the forces created due to the expansion of combustion products in the cylinder head. Piston rings confine the gases in the top of piston and control the lubrication of cylinder wall.

Recardo<sup>17</sup> and Williams<sup>18</sup> are the first propounders of corrosive failure of cylinder liner and piston rings. The greatest wear is found to occur at the top end of the topmost piston ring travel and decreases from there down. In many cases considerable wear of cylinder liners has been found at the end of lower ring travel but to a lesser extent than the top one. Incidentally friction also increases at the top and the bottom of the stroke. An ideal pattern of wear in a cylinder has been reproduced by Jominy.<sup>19</sup> However, there is a great deal of disagreement as to the cause of wear of engine liners. Many are of the opinion that corrosion is the chief cause while some others feel that failure of lubrication is the source of weakness. Influence of foreign particles and oxidation of exposed surfaces also contribute towards metal

wastage. Operating conditions play the most important part, when dealing with such a specific problem on wear.

The metallic loss from the cylinder bore is of a very fine nature. The fine particles are supposed to be (a) a formation of "Beilby layer" of broken metal detached from the bulk crystalline body having its original crystalline structure and some amorphous product bounding these crystals, (b) hard particles separating out from a soft matrix, due to shearing action or the corrosive action, if present.

These two factors combine to form an abrasion process between the liner and ring further loosening more of abrasive particles. It depends on the capacity of the lubricating oil to remove such abrasive particles and cause only normal wear of liners and rings.

According to Recardo<sup>17</sup> the wear is increased considerably when a diesel engine is started and stopped too frequently, the engine being allowed to cool in between. This is due to the moisture condensates from the combustion product, forming acidic products and corroding the cylinder walls. Increased sulphur from the fuel also causes increased wear forming  $\text{SO}_3$  and  $\text{H}_2\text{SO}_4$ . Low water jacket temperature will decrease corrosive wear. Too high a temperature due to fuel combustion accelerates wear by thinning down the lubricant. Attempts have been made to substitute stainless steel liners in place of cast iron liners without any success, thus showing that wear of cylinders and rings may not be due to corrosive action alone. Another point to be considered is the oxidation of surfaces. It has been indicated that the formation of iron oxide is a major factor determining the extent of wear, though the formation of a thick oxide film protects the surface from further attack.

The role played by a lubricant in wear prevention cannot be over emphasised. Hence it is of importance to know whether the oil performs its job adequately in cylinder and piston rings. Wear is mainly caused by the destruction of oil film on the surfaces. According to Horaik<sup>20</sup> the oil films may be destroyed due to any of the three following causes: (1) the specific pressure between the ring and cylinder attaining a very high value; (2) high temperature lowering the viscosity of the oil thus forcing the oil away even at low pressures; (3) dilution of oil reducing the viscosity; (4) presence of extraneous particles like sand, grit, etc. causing high localised pressure and dry friction. Apart from these the existing gas mixture pressure may also sometime remove the oil from the surfaces. Velocity of moving parts may also govern the stickability of lubricants. The temperature attained inside the cylinders sometimes reaches as high as  $2800^\circ\text{C}.$ , which is enough to vaporise, carbonize or burn the oil thus defeating the purpose of lubrication. Oils that are subpar show very high rates of metal pick up and hence careful selection of proper grade of oil is very important. Lastly, the design feature of scraper ring is important in that it encourages oil wedge action. Chemical analysis of lubricants and of piston deposits always shows a predominance of sulphate either in combined or free acidic state. This is one of the sources of acidic corrosion of liners and rings. Employing a basic counter agent or additive

to prevent this has met with some success. One approach is to circulate lubricating oil over a bed of lime to impart the necessary basicity.

Another factor influencing acidic corrosion is the fuel which affects piston ring wear directly. Wear is directly related to the combustion process which is in direct relation to design. Too rich a mixture or too lean a mixture of fuel also affects wear increasing it 100 times the basic wear. Some of the constituents of the fuels have indirect relation with corrosive wear. The most common fuels used in internal combustion engines are petroleum products like gasoline, kerosene, diesel oil, etc. and sewer gas. In the absence of any other corrosive agents the increase in wear can be attributed to the  $\text{CO}_2$  and  $\text{H}_2\text{O}$  formed from the combustion of hydrocarbon fuel. The corrosive action of  $\text{CO}_2$  can be avoided by higher temperature operation. Sulphur in commercial fuel forming  $\text{SO}_2$  and  $\text{SO}_3$  during combustion process also increases wear and this to some extent can be overcome by adding some alkaline earth metal to the fuel. Heavily leaded fuel also has to be avoided, since it forms deposits and causes abrasive wear. Apart from these there may be factors which play their part only in actual service conditions and for which no experimental data are available. Lastly many of the cylinder failures reported as "scuffing" are supposed to be due to "surface disintegration" or mechanical failure of liner surface. This breakdown of liner surfaces can happen even without the breakdown of the lubricating oil film.

Mechanical failure of surfaces seems to have direct relation with surface finishing operations and the ultimate surface finish. The ultimate surface finish is the result of "wearing in" or "running in" process. For a long time cylinder surfaces were finished to the profilometer range of less than 6 micro inches, resulting in the scuffing of piston rings and uneconomical use of lubricating oil. This led to the improved process of finishing by honing operations to  $60^\circ$  diamond shaped visible pattern on the surface with horizontal or concentric marks held to minimum. Of late some passenger car engines have bores that have been "Knurled". "Knurling" the bore has resulted in reduced wear performance. Further, they function as oil reservoirs and deposits for combustion products which can be removed easily.

Nonmetallic coatings as obtained by chemical treatment produce to some extent a wear resistant surface. The process consists in giving a phosphate coating by a hot dip process. This operation not only reduces scuffing but relieves the cold work strains and cleans the surface. Hardening by flame the maximum wearing surfaces or induction hardening is another method for minimising wear. Chrome plating the cylinder wall helps in inhibiting corrosive and abrasive wear. The process known as "porous chrome" plating is employed. Lastly piston rings are made 20 points lower Brinell than cylinders and liners having a V.P.N. of 530. This increases the life of cylinders by 200 per cent. Cylinder liners, rings, etc., may wear out considerably with all the above precautions taken, if no care is exercised in checking the dust brought in through the air intake. Dust particles are the deadliest enemies in any form of mating of materials.

### Metallurgical Factors

Cylinder liners are of two types: (1) wet and (2) dry. Dry liners are press fitted to cylinder block and wet liners of a very much thicker section act as a seal to the cylinder water jacket as well as to cylinder bore. Liners must be of sufficient strength to withstand operational stresses and be resistant to any form of wear. Cast iron or alloyed cast iron are used for liners with a B.H.N. ranging from 200 to 280. Heavy duty engines have liners of 300 to 550 B.H.N. They are generally centrifugally cast, thus affording a variation in microstructure from outside to inside ; this is controlled by proper designing and working schedule.

Dry liners are usually of cast iron with slight alloy additions of Ni or Cr. They are then subjected to stress relief treatment and no other heat treatment. Hardness varies from 220 to 280 B.H.N. Unhardened liner with a random distribution of "A" type graphite flake having pearlitic matrix is supposed to be the ideal microstructure of a wear resistant cast iron liner material. Ferrite primary or secondary is completely absent. The composition range common to all practices is as follows: Total carbon, 2.85-3.60 ; silicon, 1.25-2.25 ; manganese, 0.5-1.0 ; phosphorus, 0.2-0.55 ; sulphur, 0.12 max. ; chromium, 0.3-0.5 ; nickel, 1.0-1.5 ; copper, 1.0 ; and molybdenum, 0.25-0.5 per cent.

Metals on the lower range of the above composition will have a satisfactory microstructure if the casting is not allowed to chill.

According to Lane<sup>21</sup> grey iron in which graphite flake is randomly distributed is the first consideration for good wear resistance. Secondly there is an optimum size of graphite flake for any given section and thirdly the austenitic grain size governs the wearing quality of any material.

The addition of elements like Cu, Ni, Cr and Mo have been studied as to their wear resisting characteristics. When alloyed with cast iron Ni, Cr and Mo in combination show good wear resistance. Hardened cast iron liners of the following composition with a hardness of 500 B.H.N. have increased liner life to threefold.

				(%)
Total carbon	...	...	...	3.25-3.5
Si	...	...	...	2.05-2.25
P	...	...	...	0.25 max.
Ni	...	...	...	0.1 -0.25
Mn	...	...	...	0.55-0.6
Cu	...	...	...	1.0 -1.50
Cr	...	...	...	0.55-0.8

For longer life porous chromium plated liners are found to be best suited. This type of plating which is only 0.004 in. thick combats both corrosive and abrasive wear. The only limiting factor being the high cost of production. More recent practice in U.S.A. is the copper plating of the outside of cylinder liners. Copper being a good conductor accelerates the rate of heat transfer in conjunction with the water jacket.<sup>22</sup>

Piston rings have been made of various materials and the present day ones are usually of cast iron, alloyed or unalloyed in hardened or unhardened condition. Recent trend is towards making rings of 1 per cent C steel in a martensitic or troostitic condition. Yet some others use segments of cast iron and steel in combination. The best composition having good wear resistance is: Total carbon, 3.10-3.45; Si, 1.9-2.4; Mn, 0.50-0.90; P, 0.4-0.65; S, 0.10 max.; Cr, 0.40; and Ni, 0.4 per cent.<sup>22</sup>

The micro-structural requirements of a good piston ring is the same as that of the cylinder, i.e., a wholly pearlitic matrix in a phosphide network.

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# Porous Chromium Plating of Ferrous and Light Metal Alloys in relation to their Utilisation in the Manufacture of Cylinders, Pistons, and Piston Rings of Internal Combustion Engines

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The wide use of chromium plating since it was commercialised in 1924 is due to the coexistence of very desirable properties of the electrodeposit. The electroplate resists atmospheric corrosion at ordinary and high temperatures, possess a high degree of hardness and wear resistance, has a low coefficient of friction and good reflectivity. Early use of chromium plating has been mainly on the decorative side—chiefly in the manufacture of automobile and railway parts requiring pleasing external finish. Chromium plating has now been extended to engineering, employing what is known as the hard chromium plate. Gauges, tools, drawing dies, moulds for the plastic industry and generally parts requiring resistance against mechanical wear have been chrome plated thus ensuring a longer life.

It was but natural that the manufacturers of internal combustion engines should have thought of chromium plated cylinder bores for an extended life of the engine. However, piston and piston ring seizure frequently occurred in chromium plated cylinder bores ; and this led to an unfavourable reaction in chrome plating of cylinder bores. Van der Horst, a Dutch Engineer, attributed the seizure to the peculiar property of the chromium surface, namely, that it does not retain the lubricant. He ultimately succeeded in producing a porous surface on hard chromium plate. This porous surface not only helps in retaining the lubricant but also interrupts galling. It provides a place for embedding of abrasive particles and prevents the scouring of sliding surfaces. Surface disintegration of cylinders is also prevented. The value of porous chromium plating was very well recognised, and engines of planes used in North African campaign in World War II were chromium plated as a measure against sand abrasion and for salvage.

A porous chromium deposit is essentially a hard chrome plate traversed in all directions by multiple cracks. This structure is obtained by anodic etching of the hard chromium plate, the surface of which has minute cracks. The width of these cracks can be increased to desired stage by appropriate choice of working conditions of plating and etching. Two types of porosities exist—the canal and the pin-point types. These are obtainable by suitable

plating conditions. Observation has shown that there is an extensive and rapid spreading of the lubricants on porous chromium than on other surfaces, and the mechanism of spreading is taken to be of capillary nature. The spreading of the lubricant is more rapid on the canal than on the pin-point type.

Porous chromium plating was first done on cast iron cylinders and is now mainly restricted to the bore of big diesel engines—especially marine and locomotive types. Life of chromium plated aero engine steel cylinders was found to be almost doubled. Plating of small bores of high speed diesel and petrol engines is considered to be too expensive, and chromium plating of top piston rings is considered to be quite effective in reducing cylinder wear which is about a third of that observed with unplated rings. The porous surface on the rings permits them to seat quickly without getting worn out and condition the cylinder during the seating period. The life of the ring has been increased to almost five times the normal life of an unplated ring. The extra cost incurred in chrome plating has been well justified by the longer life of cylinders and piston rings.

The use of aluminium alloys in the construction of the cylinder had to remain inoperative due to the poor resistance of the alloy to the cast iron piston rings. An effective way by which the poor sliding characteristic of aluminium alloy has been attempted to be rectified is by covering the sliding surface with a leached or metal layer. Of the latter, chromium plate has been most effective because of its very desirable properties. It has been observed that in comparison with cast iron cylinders chromium plated light metal cylinders show reduced piston temperatures particularly in the ring zone ; cylinder and cylinder-head temperatures are more nearly equal and the danger of distortion is reduced. Further endurance tests have indicated the relative insensitivity of chromium plated aluminium cylinders to piston-sticking, which tests if conducted on cast iron cylinders would have led to their destruction.

The different steps involved in the production of porous chromium surface are as follows: (a) initial dimensional modification for taking the chromium deposit and check up of the same ; (b) pretreatment of the part so as to obtain such a clean surface as will facilitate the formation of a firmly adhering chromium deposit ; (c) correct jigging, and well controlled plating conditions involving the regulation of temperature, voltage, current density, electrolyte concentration, etc., (d) anodic etching to secure porosity, the duration being adjusted so that the etching is restricted to a short depth on the chromium plate only, and never to reach the basis metal ; (e) final finishing and dimensional check. Naturally, different materials used for cylinder, piston and piston ring and different shapes provide individual problems for solution.

In general, i.e., either for light metals or for ferrous alloys, the electrolytic bath has been of the chromic acid-sulphuric acid type of ratio 100:1. The operating temperatures range from 50-55°C. ; the current density varying from 35-50/dmsq. The voltage is round about six volts. Another type of bath containing fluosilicic acid as the catalyst has also been used with success.

In a series of experiments conducted with steel it has been found that in order to obtain a smooth hard chromium plate, one essential feature is good

surface finish preparatory to plating. In this connection it has been found that a surface obtained by electropolishing has been superior to hand or machine polishing. Further, it has also been found that the tedious grinding of chromium plated valves can be eliminated by anodic polishing. Thus the role of electropolishing as preparatory and finishing steps in porous chromium plating is significant, and well worth studying in the making of porous chromium bores and piston rings.

## Concluding Session

The concluding session was held in the Department of Internal Combustion Engineering on 6 April 1952 at 4 p.m. Dr. J. C. Ghosh, Chairman of the Internal Combustion Engines Research Committee, Council of Scientific and Industrial Research, was in the chair.

Prof. M. S. Thacker initiated a discussion on technical aid to industry by the laboratories. A large number of guests participated in the discussion. Representatives of the industry who were present very warmly welcomed the assurances of Dr. J. C. Ghosh and Prof. M. S. Thacker that the laboratories in Bangalore would give all the assistance that industry may need.

